AGRICULTURAL PROCESS ENGINEERING

THE FERGUSON FOUNDATION
AGRICULTURAL ENGINEERING SERIES

AGRICULTURAL PROCESS ENGINEERING

S. M. Henderson and R. L. Perry

TRACTORS AND THEIR POWER UNITS

E. L. Barger, W. M. Carleton,
E. G. McKibben, and Roy Bainer

FARM STRUCTURES

H. J. Barre and L. L. Sammet

AGRICULTURAL PROCESS ENGINEERING

S. M. Henderson
ASSOCIATE PROFESSOR OF
AGRICULTURAL ENGINEERING

R. L. Perry
PROFESSOR OF
AGRICULTURAL ENGINEERING

UNIVERSITY OF CALIFORNIA

1955
JOHN WILEY & SONS, INC., NEW YORK
CHAPMAN & HALL, LTD., LONDON

Library of Congress Catalog Card Number: 54-12684

PRINTED IN THE UNITED STATES OF AMERICA

Preface

Agricultural processing is defined as any processing activity that is or can be done on the farm or by local enterprises in which the farmer has an active interest. More specifically, any farm or local activity that maintains or raises the quality or changes the form or characteristics of a farm product may be considered as processing. Processing activities are undertaken to provide a greater yield from a raw farm product by increasing the amount of finished product, the number of products, or both, and to improve the net economic value of a product by raising its quality or the yield or by decreasing the cost of production.

A few agricultural processing activities are:

Cleaning, sorting, grading, treating grain, seed, nuts, cotton, fruits, vegetables, peanuts, eggs.

Drying or dehydrating grain, seed, forage, nuts, tobacco, fruit, vegetables, milk, hops, eggs.

Grinding and mixing animal feeds, fertilizers.

Milling sorghum, sugar cane, rice.

Canning fruit and vegetables.

Packing fruit and vegetables.

Dressing meat and poultry.

Freezing fruit, vegetables, meat.

Conditioned storage and transportation of products.

Other processing, such as pertaining to fluid milk, butter, cheese, ice cream, honey, molasses, mint, turpentine, fiber crops.

A processing job consists of a series of events or "unit operations." Many of the unit operations are used in more than one processing job, materials handling, cleaning and sorting, drying, for example. Many devices or procedures that are not treated adequately in the average agricultural engineering curriculum are important in agricultural processing; fans, heat transfer, instrumentation, work simplification, are examples. The unit operations, processes, devices, and procedures that appear to us to be most important in agricultural processing are:

- Size reduction
- Cleaning and sorting
- Drying and dehydration
- Concentration by evaporation
- Refrigeration
- Mixing
- Materials handling
- Air conditioning
- Steam generation and use
- Heat transfer
- Pumps and fans
- Plant layout
- Work simplification
- Instrumentation

This textbook was designed primarily to assist in teaching the engineering elements of agricultural processing to advanced students in agricultural engineering. We have assumed that the student will have completed courses in calculus, thermodynamics, and perhaps heat-power engineering. Although most students will have completed courses in fluid mechanics, a chapter is included for review, reference when working problems, background to the chapter on fluid-flow measurements, and a source of information on fluid flow through porous media. Likewise, many students will have completed work in economics dealing with agricultural costs. Nevertheless, we have included a chapter on cost analysis which we believe sets out a procedure for analyzing the cost of an operation which will be most valuable to an engineer. The text material was prepared from the basic standpoint where possible. Unfortunately, certain subjects treated (cleaning and sorting, plant layout, for example) are not sufficiently developed to treat rigorously. Furthermore, other subjects were treated descriptively because we felt that the agricultural engineer would be more interested in applications than design.

We recommend that the instructor compose his course of those unit operations common to the processing in his region, thus fitting his students to handle more jobs than would be possible if the course were set up on an enterprise basis. Extensive use of the reference material and the recognized engineering handbooks is recommended. The problem sets can be supplemented with problems typical of the local area and the interests of the students. The appendix was prepared to assist with problem solution, although it can be used as a reference source by the practicing engineer.

We are happy to acknowledge the many contributions by those in industry, university, and Federal work in the preparation of the manuscript. Space does not permit a complete listing of

those who assisted in some significant manner. We wish to express our appreciation for important contributions by the following: Dr. R. M. Barnes, University of California (Los Angeles); Prof. J. C. Hempstead, Iowa State College; Dr. R. G. Folsom, University of Michigan, Ann Arbor, Mich.; Ronald Banton, A. T. Farrell and Co.; Hill Shepardson, Hart Carter Co.; A. J. Bouey, Westinghouse Corp.; Gilbert T. Bowman, Pittsburgh Equitable Meter Division, Rockwell Manufacturing Co.; Frank Maytham, Link-Belt Co.; E. C. Meyer, Minneapolis-Honeywell Regulator Co.; and Waldo H. Kliever, Clevite Brush Development Co.

We are particularly indebted to The Ferguson Foundation for the material support that permitted the manuscript to be prepared and to Mr. Harold E. Pinches, representing the Foundation, for his encouragement and counsel. We further wish to specially recognize the University of California (Davis) for cooperating in the development of the project and for supplying many of the material facilities.

S. M. Henderson
R. L. Perry

December, 1954

Contents

CHAPTER 1

The Engineering Approach

Engineering has been defined as "the art and science of utilizing the forces and materials of nature for the benefit of man and the direction of man's activities toward this end."

The above definition implies the division of engineering into two activities: (1) art and (2) science. Engineering is based upon the pure sciences, physics, chemistry, mathematics, which produce the same result under a certain set of conditions irrespective of when or where these conditions exist. The amount of energy required to melt a pound of ice, the velocity of sound at standard conditions, the amount of air required to burn a pound of ethyl alcohol and the amount of heat produced by this means, the force required to compress a spring 0.376 in. when the spring data are known, the time required to empty a water tank when the orifice constant is known, and the product of 7068 and 386 are reproducible irrespective of the source of the data or the extraneous conditions.

However, just as soon as we begin to apply standard values to natural materials and situations we note a variation in the results which may be related to time, location, or other conditions. The spring mentioned above after a few months' use may compress more than 0.376 in. when tested, because of fatigue. The sample of commercial ethyl alcohol might produce 12,950 Btu per lb rather than 13,170 Btu as expected. The above-mentioned orifice constant can be secured from published charts and tables, but it is improbable that the indicated value would be exactly representative of the particular orifice under consideration. Consequently, in most engineering calculations, the result is not exact. In general work, a variation of 2 per cent is accepted.

Many engineering calculations are rational in concept but empirical in application because of an important factor or factors

which must be determined experimentally. For example, the rate of drying of agricultural products can be expressed thus:

$$dq/dt = kp_s(p_m - p_a)$$

The term k, which would probably be called a constant, but is not, must be determined experimentally for each material. It would not apply above a certain moisture content which would be the dividing point between combined and free moisture. Since there is an overlapping between them, no exact point exists. Furthermore, the value will vary, owing to the weather and soil conditions under which the product was grown, the variety, its treatment between harvest and the time drying is started, etc. p_m is the vapor pressure of the material and is taken from an equilibrium moisture curve which has previously been determined by observation and which is subject to the same variation, more or less, as the so-called constant k. Although the vapor pressure of moisture in the air p_a and the saturated pressure p_s are also empirical, values are reliable.

The performance of a wood member under a load can be calculated on the basis of certain rational formulas that yield tension in the outer fiber, maximum horizontal shear of the member, and the amount of flexure. However, these calculations require certain "constants" that define the limits of performance, ultimate strength, elastic limit, and modulus of elasticity. These constants are averages of a great number of individual observations which may vary considerably. Consequently, since any single member may be much weaker than the average, a factor of safety of 2 to 6 is applied to the rational calculation to insure satisfactory performance.

Certain engineering relationships can be expressed graphically or mathematically even though the basis for the relationship is not known or is not apparent. This type of relationship is based entirely on experimental data and is completely empirical. Examples are the power-particle size relationship for grinding grain, change of viscosity with temperature, resistance of a barn of hay to air flow, and the pressure-discharge relationships of a centrifugal pump.

The *science* of engineering is that phase of the field which is exact and rational. It is exact, and for any set of conditions the end point will always be the same. The conditions can be related

mathematically and are based upon laws that can be rationalized upon the pure sciences. Any constants or variables that must be determined experimentally can be defined and do not vary greatly after being established.

The *art* of engineering refers to the ability to judge, estimate, and manipulate the uncertainties of engineering to a satisfactory solution of a problem. It refers to a procedure that has been found by a series of trial and error events, carried out in as logical a sequence as possible, to produce a desired result without knowledge of the basic principles involved. It refers to the use of empiricals in an efficient manner. The Chinese made iron and steel, and the Egyptians glass; although the Chinese knew nothing of metallurgy, and the Egyptians nothing of the science of glass making. The Indians fertilized their corn with dead fish, but they knew nothing of plant and soil science. Portland cement and petroleum products are well developed, but the chemistry of neither is completely known. Farm crop driers are designed for satisfactory performance, although little is known about the drying characteristics of the materials except in an over-all way.

The field of farm-products processing contains more engineering uncertainties than the more common engineering fields. Successful treatment of a problem frequently requires that the engineer estimate, extrapolate, or secure information empirically to solve a problem. Occasionally, decisions must be based upon intuitive judgment. This procedure is hazardous but sometimes necessary.

It is this ability, the ability to evaluate the uncertainties, that differentiates an engineer from a pure scientist, and the engineer's success will depend in great measure on the skill with which he handles these uncertainties.

EVALUATING THE UNCERTAINTIES

There is no definite set rule or procedure for evaluating the uncertainties. If there were, they would not be uncertainties. However, a few helpful procedures, factors, and principles can be given as follows:

The Idealized Situation. An engineering problem or project, in design, development, or research, can best be evaluated by establishing all known facts and procedures which are or appear to be related to it. If the problem is first idealized on the basis

of known rules, factors, and laws, it will serve as a standard or measure of fit or performance of the final engineering decision.

For example, the amount of heat energy needed to reduce the moisture content of a ton of grain from 24 per cent to 14 per cent must be determined or estimated accurately. In making this reduction, 234 lb of water would have to be removed. Now, no data are available showing the exact amount of heat energy required to effect this reduction. However, if the water were removed by vaporization, approximately 234,000 Btu would be required. This would be considered the idealization. We know that most of the moisture already exists in a state other than liquid. Consequently, it is probable that less energy would be required than the above figure indicates. But, on the other hand, the moisture at the center of each kernel must be moved from the center to the surface, a procedure that will require additional energy. In the absence of explicit data, we assume that the energy required to move the moisture from the center to the surface equals the reduction resulting from the presence of the moisture in a nonliquid form. Therefore, we assume that the vaporization figure applies.

Variation. The engineers' factor of safety is needed for two reasons: (1) insufficient or incomplete basic information and/or (2) inability to forecast future conditions related to the operation. The variations in products, weather, markets, demand, etc., which affect many of the engineering aspects of a problem, are difficult and sometimes impossible to evaluate.

A knowledge of statistical procedures will aid in providing a satisfactory answer to many problems involving variable or uncertain factors. It is especially helpful for those engaged in research who are attempting to establish basic relationships.

Statistics, especially analytical statistics, may be defined as the mathematical science of variation. The procedures that it embraces may be used to (*a*) show a mass of data in an easily understandable graphical form, (*b*) resolve the data into a mathematical formula, or (*c*) determine its reliability. The statistical evaluation of reliability is very important since it aids in determining the qualitative value of data, the probability of certain events, and the number and characteristics of samples that must be taken to yield significant results. It would be impossible to

give the reader a workable knowledge of statistics in a few short paragraphs, but his interest can be excited by the following example.

A class of 20 students was divided into two sections of 10 students each who were taught by different instructors. The final grades for the two sections were as tabulated.

	Section A		*Section B*
	78		82
	90		76
	60		81
	77		97
	84		84
	87		99
	89		73
	96		78
	72		86
	54		87
Average	78.7	Average	84.3

The instructor in section A was criticised for doing a poor job of teaching because (1) the class average was lower than that of B, (2) the poorest student in A made a lower grade than the poorest student in B, and (3) the best student in A had a lower grade than the best in B.

However, a statistical analysis showed the following. The standard deviation, which is a measure of variation, was found to be 8.4 for section A and 8.5 for B. This finding indicates that two-thirds of all the possible grades represented by the sample would fall within the range defined by the average plus and minus the standard deviation. In section A this range would be 70.3–87.1 and in B, 75.8–92.8. Note that of the total range, 70.3–92.8 of both sections, 75.8–87.1 or 50 per cent of the entire range is common to both. This indicates that it is possible that the grades are a chance randomization of a single group rather than a result of poor teaching. A standard statistical test of significance applying the pooled standard deviation, 8.5 and the difference between the averages 5.6, shows that there is only one chance out of 20 that the section difference is due to poor teaching. The difference between the averages would have to be over 6 before the statistician would consider the possibility that the result was a function of difference in teaching ability.

This simple example demonstrates in an elementary way the difficulties that may arise when data are taken at face value. The engineer should be analytically critical of the value of a number which is an average of a series of observations. What is the variation in the observations from which the average is derived? Is the variation due to the method of sampling? Are the observations comparable? What causes the individual observations to be inconsistent? And finally, just how accurate or representative is the average? The standard deviation as noted before is an index of variation or accuracy. Similar indices are available for treating a series of comparable averages, for evaluating the fit of a curve to plotted data or data to a curve, etc.

A useful approximate relationship to remember is that in a normally distributed sample,

$$(\text{Range of means}) = (\text{Sample range})/\sqrt{\text{No. of samples}}$$

For example, the percentages of the moisture content of 5 samples of grain taken from a field are respectively 11, 17, 15, 15, and 13, the average being 14.2 per cent. The range of the means or range within which the true average probably exists is

$$(17\text{–}11)/\sqrt{5}$$

or 2.68. If the above 5 samples are true random samples, the probability is approximately 2:3 that the true average will fall between $14.2 \pm 2.68/2$, or 12.9 to 15.5 per cent.

This statistical discussion is not intended to provide the reader with a tool for accurate evaluation of a varying situation but is intended to excite interest and caution and to indicate that factors that vary considerably may yield finite results when treated properly.

Economics. The economic phase of an engineering problem (discussed in detail in Chap. 13) must never be overlooked. Many engineering processes are designed specially to reduce production costs, usually by speeding up the process, eliminating or making manual labor more efficient, or reducing overhead costs. A new or improved engineering procedure must always be judged by its economic value. The effect may be indirect in that a particular machine or operation may contribute to better application of another unit.

In processing work there is usually a distinct if not small difference between the cost of the raw products and the selling price. The processing operation must be performed well within this economic bracket if a fair return on the investment is to be assured. Economic improvement of an operation is usually produced in one of two ways: by reducing the cost of production per unit or by raising the net return per unit. Increased net return could result from reducing the salvage, using the by-products more effectively, or raising the quality of the product. Although the processing engineer may not be conscious of it, his activities are usually directed toward one of the above mentioned objectives.

CHAPTER 2

Fluid Mechanics

NOMENCLATURE

A = area, sq ft.
A_f = wall-effect factor, dimensionless.
C = clearance, ft.
D = diameter.
E = number of rows of tubes normal to fluid stream.
F = friction loss, ft lb per lb or ft.
f = coefficient, dimensionless.
G = flow rate, gal per min.
g = acceleration of gravity, 32.2 ft per sec^2.
h = height, ft.
K = a proportionality constant.
L = depth, ft.
l = length, ft.
m, n = exponents.
P = force, lb.
p = pressure, lb per sq in.
p' = pressure drop, in. of water.
R = hydraulic radius, ft.
Re = Reynolds number, dimensionless.
t = time, sec.
V = velocity, ft per unit of time.
V_0 = air rate, cu ft per min sq ft.
v = void space, a decimal.
W = work energy, ft lb per lb or work head, ft.
w = weight rate, lb per unit of time.
x = a quantity of fluid, lb.
y = separation distance, ft.
ϵ = roughness factor, dimensionless.
γ = specific weight, lb per cu ft.
μ = fluid viscosity, lb per ft sec.
μ_f = fluid viscosity, lb-sec per ft^2.

A complete study of fluid mechanics would be divided into two parts: fluids at rest or hydrostatics and fluids in motion or hydrodynamics. The first part, that treating of fluids at rest, will be

assumed to have been covered in the required physics, chemistry, or basic-engineering courses. The second part, which deals with the various factors affecting the relationship between the rate of flow and the various pressures tending to cause or inhibit flow, will be treated in detail. More specifically, such things as the amount of a fluid, water, air, milk, or brine, e.g., flowing through a system of pipes if the pressure causing the flow is known or the power required to produce a desired rate of air flow through grain which is to be dried will be considered. The individual factors involved will be studied and related to the various fluid-flow applications in which the processing engineer is interested.

BASIC CONSIDERATIONS

2.1. Classification of Fluids. Fluids are classified as either compressible, gases; or incompressible, liquids. Liquids are compressible to a very small degree, but no significant error results in most engineering calculations if incompressibility is assumed. The principles of fluid flow apply equally well in both cases.

2.2. Analytical Basis. The analysis of any fluid system must take into consideration one or more of the following:

1. Conservation of mass.
2. Conservation of energy.
3. Newton's laws of motion.
   a. Every body continues in a state of rest or of uniform motion in a straight line unless compelled by force to change that state.
   b. The rate of change of momentum is proportional to the force applied and takes place in the direction of the force application.
   c. To every action there is always an equal and opposite reaction.

The term fluid system as herein considered refers to any part of a building or unit or series of units of equipment which is related to fluid mechanics. It may be a complete system such as a water or ventilating system for a processing plant or a single unit such as a valve, filter, pump, or a length of pipe. It must have definite boundaries. Consider Fig. 2.1. This hydraulic system consists of a pump, filter, valve, elbow, and connecting

pipe and is only a small part of a complete system. Points A and B are the boundaries that define the system under consideration.

The simplest case is based on the assumption that all conditions are constant with time at each point in the system. Fluids frequently flow with an irregular rate, that is, surge, under certain conditions. Situations where this condition must be recognized are few.

If the rate of flow is constant at any point and there is no accumulation or depletion of fluid within the system, the mass rate of flow at any number of points within the system must be constant since matter can be neither created nor destroyed. The mathematical statement of this follows:

$$A_1V_1\gamma_1 = A_2V_2\gamma_2 = \cdots = A_nV_n\gamma_n = w \tag{2.1}$$

where A = cross-sectional area of conduit, sq ft.
V = linear velocity of fluid, ft per sec.
γ = specific weight, lb per cu ft.
w = weight of material flowing, lb per sec.

In engineering, a four-dimensional system including force, mass, length, and time is most generally used. The pound is used for the unit of mass (quantity of matter). The pound is also used for the unit of force. This practice has developed because the quantity of matter is measured by observing the force which is exerted on a balance or scale. Thus when we speak of weight, we commonly refer to the mass (quantity of matter) rather than the force (earth-pull) which actuates the scales.

In general, force is proportional to the product of mass and acceleration.

In the engineering dimension system, the proportionality constant is $1/g_c$, where $g_c = 32.17$ (lb mass per lb force)/(ft per sec^2). Note that this is not simply g, the gravitational acceleration, which has the dimensions of feet per second squared. Thus

$$F = ma/g_c$$

Throughout the text, the expression mass is intentionally avoided because of its common connotation w/g, that is, Weight/Gravitational acceleration. The variation of weight of a given quantity of matter with geographical location is so small as to be overlooked in agricultural processing. When weight (strictly speaking, earth pull) is used to designate quantity of matter, one unconsciously multiplies by (pounds of matter per pound of weight). In using weight, the relation between force and mass becomes

$$F = \frac{w(\text{lb mass per lb force})a}{g_c} = \frac{wa}{g}$$

Example. Water is flowing in a pipe 6 in. in inside diameter at a velocity of 60 ft per min. The pipe enlarges to 12 in. in diameter. What is the velocity in the larger section and the quantity flowing?

$$\text{Weight rate of flow} = \text{Area} \times \text{Velocity} \times \text{Lb per cu ft} = w$$

$$= \pi(\tfrac{3}{12})^2 \times 60 \times 62.4 = 736 \text{ lb per min}$$

The velocity in the larger section

$$= \frac{736}{\gamma \times A}$$

$$= \frac{736}{62.4 \times \pi(\frac{6}{12})^2} = 15 \text{ ft per min}$$

(Note that for liquids and gases, where the change in density is negligible, the velocity varies inversely as the square of the diameter.)

A useful equation is

$$V = 24.5G/D^2 \tag{2.2}$$

in which V = velocity, ft per min.
G = quantity flowing, gal per min.
D = pipe diameter, in.

Likewise, since energy can be neither created nor destroyed, the total energy represented at one point in the system must equal that at any other point plus intervening transfers. This condition is the basis of all hydrodynamic calculations and will be treated in considerable detail.

MECHANICAL ENERGY BALANCE

Consider the hydraulic system shown in Fig. 2.1 located above a reference plane which might be represented by a level floor and which is defined as existing between points A and B. The total mechanical energy involved in this or any other system is made up of three elements.

1. Energy available because of elevation above a reference plane.
2. Energy available because of internal pressure.
3. Energy available from the moving fluid.

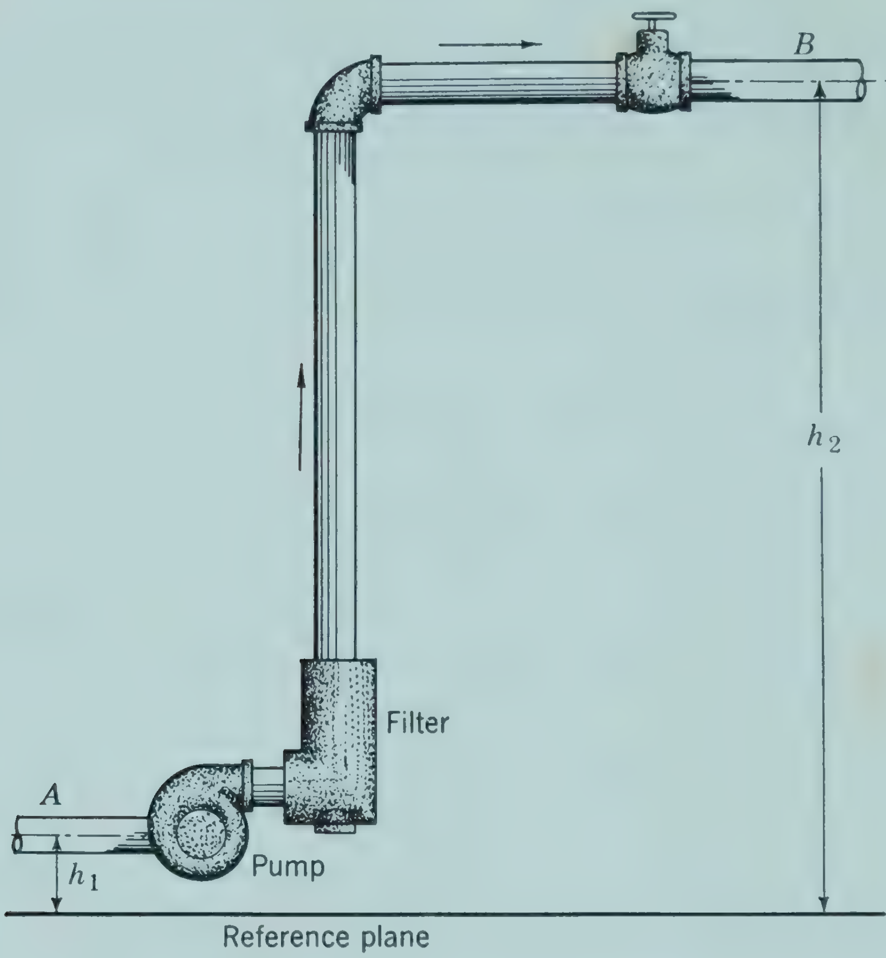


Fig 2.1. A hydraulic system.

2.3. Elevation Energy. A quantity of fluid of weight x is considered flowing through the system. At point A it has a potential or elevation energy value of

$$xh_1 \tag{2.3}$$

in which x = lb of fluid.

h_1 = distance above reference plane, ft.

If the unit of fluid under consideration is released and is permitted to fall or move from its initial position to the reference plane, it has the ability to do an amount of work equal to xh; or, an amount of work equal to xh would be required to lift it from the reference plane to a point h ft above the plane.

2.4. Pressure Energy. The fluid at point A is subjected to an internal static pressure of p expressed in lb per sq in. This is in addition to the energy resulting from elevation xh and may result from a pump, elevated supply tank or other source. A quantity of potential energy exists since the x quantity of fluid must be moved past point A against this pressure. If released,

this energy is available to do work that would be defined in terms of force and distance thus:

The distance through which the force acts is,

$$x/\gamma A$$

A being the area of the conduit in square feet. The force is the unit pressure times the area or

$$144pA$$

The potential energy is the product of the force times the distance, or

$$(x/\gamma A)144pA = 144xp/\gamma \tag{2.4}$$

where p = pressure, lb per sq in.
γ = specific weight, lb per cu ft.

2.5. Velocity Energy. A body in motion possesses an amount of kinetic energy which in this case is equal to

$$x(V^2/2g) \tag{2.5}$$

where V = linear velocity, ft per sec.
g = acceleration of gravity, 32.2 ft per sec^2.

Because of this motion, the quantity of fluid under consideration if brought to rest is able to do an amount of work equal to equation 2.5, or, conversely, the same amount of work is required to bring the fluid from zero to V velocity.

2.6. Total Hydraulic Energy. The sum of the three types of energy present at A, equations 2.3, 2.4, 2.5, is the total mechanical energy available at A. This energy plus the energy W supplied by the pump less that lost because of fluid friction F in the pipes, joints, etc., must equal that present at point B because of the conservation of energy. This sum is:

$$xh_1 + \frac{144xp_1}{\gamma} + \frac{xV_1^2}{2g} + xW - xF = xh_2 + \frac{144xp_2}{\gamma} + \frac{xV_2^2}{2g} \tag{2.6}$$

Since x is common to all terms, it cancels, and the final equation is

$$h_1 + \frac{144p_1}{\gamma} + \frac{V_1^2}{2g} + W - F = h_2 + \frac{144p_2}{\gamma} + \frac{V_2^2}{2g} \tag{2.7}$$

Each term represents a head expressed in foot-pounds of energy per pound of the fluid flowing. These terms are called elevation, pressure, velocity, work, and friction heads, respectively. They may be considered as applying to 1 lb of the fluid under study, in order to facilitate the solution of a problem. The frictional-loss term, foot-pound of energy per pound fluid, is a result of friction between the fluid and the retaining walls, valves, bends, etc. and of friction and turbulence within the fluid itself. Thus results the negative sign that accompanies the frictional-loss term. The friction work is converted into heat.

The W term is positive if work is done on the fluid between points 1 and 2. If the fluid does work between the boundary references, W is negative. This relationship presupposes adiabatic flow, that is, no heat exchange takes place with the surroundings.

The energy relationship shown in equation 2.7, which results from work by Daniel Bernoulli in 1738, is known as the Bernoulli theorem. It is the basis of all fluid-flow calculations and, consequently, must be thoroughly understood by the student.

Example 1. A storage tank is located 90 ft above a body of water. Assume no energy loss in the connecting pipe. If the pipe is 2 in. in inside diameter, what would the horsepower requirement be to pump water into the tank at a rate of 500 gal per min?

Solution.

$$500 \text{ gal per min} = 66.7 \text{ cu ft per min} = 1.11 \text{ cu ft per sec} = 69.4 \text{ lb per sec}$$

The cross-sectional area of the pipe is 0.0216 sq ft. The velocity in the pipe at the point of discharge is $1.11/0.0216 = 51.4$ ft per sec.

Consider the level of the stream as the reference point.

The velocity of the water at the source is zero; the $V_1^2/2g$ term drops out. Since the pressure at both reference points is atmospheric, the $144p/\gamma$ terms cancel. The Bernoulli equation for the energy per pound of water elevated to the point of discharge is

$$W = 90 + \frac{51.4^2}{2 \times 32.2} = 90 + 41.1 = 131.1 \text{ ft water}$$

The work head is 131.1 ft of water, or 131.1 ft-lb of work would be required per pound of water to elevate it. The power required is

$$131.1 \text{ ft-lb} \times 69.4 \text{ lb water per sec} = 9100 \text{ ft-lb per sec}$$

$$\tfrac{9100}{550} = 16.5 \text{ hp required}$$

Note that 41.1/131.1 or approximately ⅓ of the power required is used to bring the water up to flowing velocity. This power could be reduced by using a pipe of larger diameter.

Example 2. Water is flowing from a pipe 4 in. in inside diameter into a pipe 1 in. in inside diameter at a rate of 200 gal per min. If the pressure in the 4-in. pipe is 40 lb per sq in. what is the pressure in the small pipe assuming no energy loss at the joint?

Solution.

The elements of the Bernoulli equation which apply are:

$$144\,\frac{p_1}{\gamma} + \frac{V_1^2}{2g} = 144\,\frac{p_2}{\gamma} + \frac{V_2^2}{2g}$$

$$A_1 = \pi\left(\tfrac{2}{12}\right)^2 = 0.0874 \text{ sq ft}$$

$$V_1 = \frac{200}{0.0874 \times 60 \times 7.5} = 5.1 \text{ ft per sec}$$

$$\frac{V_2}{V_1} = \frac{(\pi/4)D_1^2}{(\pi/4)D_2^2} = \frac{4^2}{1^2} = 16$$

$$\therefore V_2 = 16V_1 = 16 \times 5.1 = 81.5 \text{ ft per sec}$$

$$\therefore 144\,\frac{40}{62.4} + \frac{5.12}{64.4} = 144\,\frac{p_2}{62.4} + \frac{81.5^2}{64.4}$$

from which $$p_2 = -4.6 \text{ lb per sq in.}$$

Since the sign of p_2 is negative, the pressure is below atmospheric or is vacuum.

The Bernoulli equation is easy to apply to any problem if a few rules or significant points are kept in mind. The terms that apply to the problem should be carefully isolated, and those that do not apply or cancel should be discarded. Be sure that the values are expressed in the proper terms. V must be used as feet per second. A common error is failure to convert velocity from feet per minute to feet per second. When work is a factor, be sure the sign is correct.

CHARACTERISTICS OF FLUID FLOW

The manner in which a fluid flows through a system is dependent upon the characteristics of the fluid, the size, shape, and condition of the inside surface of the pipe or tube, and the fluid velocity. The frictional resistance F in the Bernoulli equation and the performance of devices for measuring flow rates which are discussed in Chap. 3 are intimately related to the characteristics of flow.

2.7. Streamlined and Turbulent Flow. In streamlined flow the fluid moves in parallel elements, the direction of motion of each element being parallel to that of any other element. The velocity of any element is constant but not necessarily the same as that of an adjacent element.

In turbulent flow the fluid moves in elemental swirls or eddies, both velocity and direction of each element changing with time. A violent mixing results, whereas there is no significant mixing in the case of streamlined flow.

2.8. Distribution of Velocities. A velocity traverse of a fluid (liquid or gas) flowing in a pipe will show that the velocity is

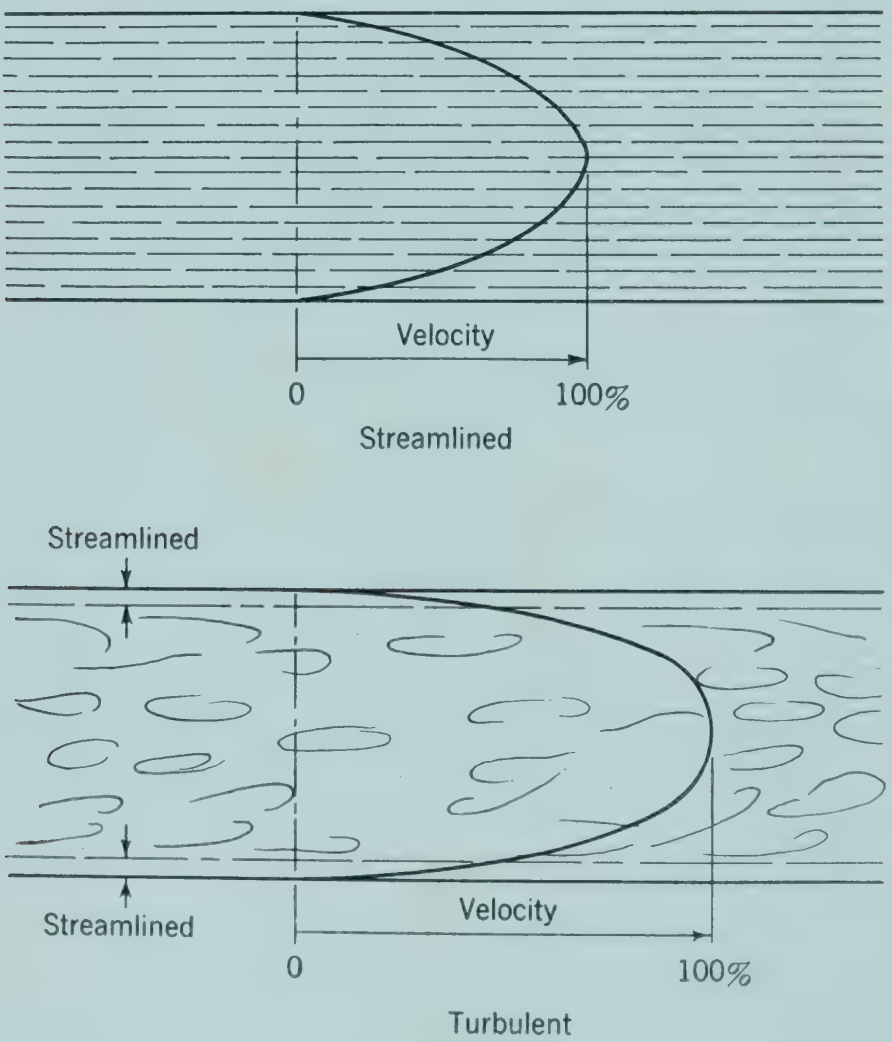


Fig. 2.2. Streamlined and turbulent flow.

highest at the center and decreases toward the surface of the container, the velocity at the surface being zero. This characteristic, which holds for both streamlined and turbulent flow, is shown in Fig. 2.2.

The velocity gradient for streamlined flow in a long circular conduit is parabolic in shape; and the average velocity is one half the maximum, which is at the center. For turbulent flow, the gradient flattens and the relationship between the maximum and average velocity changes, its exact value being a function of a number of conditions under which flow results.

2.9. Reynolds Number. Reynolds, an English investigator who was the first to demonstrate the finite existence of stream-

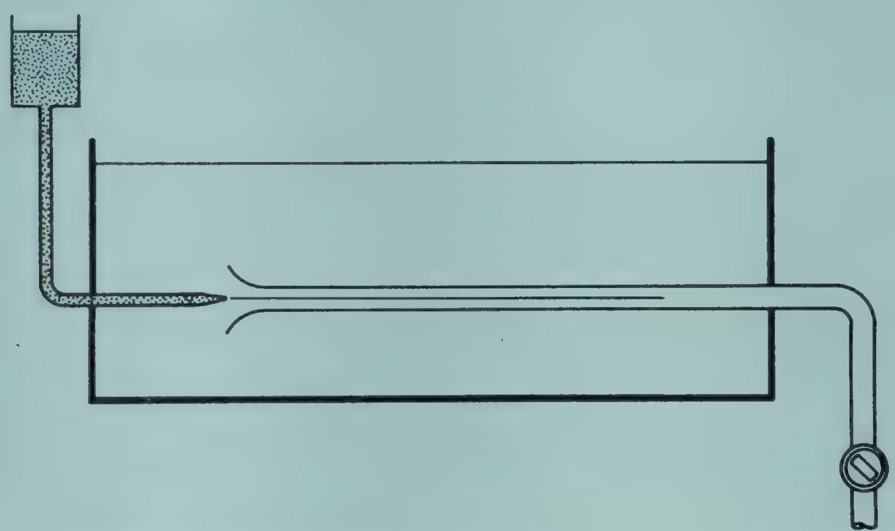


Fig. 2.3. Reynolds device for studying the transition from streamlined to turbulent flow.

lined and turbulent flow, developed the mathematical relationship defining the conditions at which flow changes from streamlined to turbulent. Reynolds introduced a thin stream of colored liquid into the bell inlet of a pipe as shown in Fig. 2.3. He found that the colored thread persisted under low velocities but as the velocity was increased there was a definite point at which the thread broke and the coloring filled the tube due to eddies or turbulent flow. The velocity at which transition results is called the *critical* velocity. Reynolds found there were four factors that affect the critical velocity. These factors and their mathematical relationship follow:

$$Re = DV\gamma/\mu \tag{2.8}$$

where Re = Reynolds number, dimensionless.
D = inside diameter of pipe, ft.
V = average velocity, ft per sec.
γ = specific weight, lb per cu ft.
μ = fluid viscosity, lb per ft sec.

Since Reynolds' time additional work has been done on flow characteristics, and it has been found that if Re is less than 2130, flow will be streamlined and, if over 4000, turbulent. For values between 2130 and 4000, the characteristics of flow will depend upon the details of the structure and any definite prediction is impossible. The above conditions hold for straight circular pipe with isothermal flow.

The above discussion considered only circular pipes. The equation for Reynolds number (2.8) can be used satisfactorily for rectangular and other shaped conduits by introducing the hydraulic radius R, which is defined thus:

$$R = \frac{\text{Area of cross section}}{\text{Wetted perimeter of cross section}} \tag{2.9}$$

For a conduit filled with a gas or completely filled with a liquid, the complete perimeter is used. If the conduit, a flume for example, is only partially filled, only the "wetted" portion of the perimeter, that contacting the liquid, is used. R for a circular pipe is

$$R = (\pi r^2/2\pi r) = r/2 = D/4 \tag{2.10}$$

from which $D = 4R$. Substituting in equation 2.8,

$$Re = 4RV\gamma/\mu \tag{2.11}$$

Equation 2.11 can be used with fair results for turbulent flow but should not be used under streamlined conditions except for nearly square or nearly circular ducts.

2.10. Viscosity. Fluid viscosity μ in equation 2.8 refers to the internal resistance of fluids to shear. The coefficient may be considered as the coefficient of friction of fluid on fluid. The latter consideration is not strictly true since one fluid layer does not actually move over another, but the analogy will serve to give the reader a physical concept of the meaning of viscosity.

2.11. Dimensions of Viscosity. Consider two layers of fluid y feet apart, the inner space being filled with fluid, as shown in Fig. 2.4. Because of the resistance to motion offered by the fluid, a force P is required to maintain a constant velocity V of the top layer relative to the lower layer. Experimental results have shown that for most fluids the required force is directly propor-

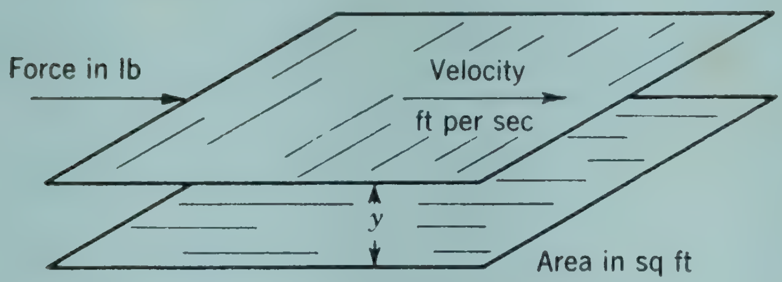


Fig. 2.4. Viscosity elements visualized.

tional to the resulting velocity, directly proportional to the area A, and inversely proportional to the separation distance y. Stated mathematically, this is

$$P = \mu_f(VA/y) \tag{2.12}$$

μ_f being a constant of proportionality which is the coefficient of viscosity. μ_f by solution is found to be

$$\mu_f = \frac{Py}{VA} \tag{2.13}$$

where P = force, lb.
y = separation distance, ft.
V = relative velocity, ft per sec.
A = plate area, sq ft.

μ_f will be found to have the dimensions, $\dfrac{\text{Lb-sec}}{\text{Ft}^2}$.

If y, V, and A are considered to have unit values, that is, one, the viscosity will be numerically equal to P and will have the dimension lb-sec per ft^2. In engineering the so-called mass viscosity μ is more commonly employed. This is obtained by multiplying μ_f by the force-mass proportionality constant g_c. Thus $\mu = \mu_f g_c$ and has the dimensions lb/sec-ft. A list of viscosities that will be useful in fluid flow calculations will be found in Table 2.1.

Published values of viscosity are usually in *centipoises* (0.01 dyne-sec per sq cm or 0.01 gm per cm-sec), the cgs unit of absolute viscosity. They must be converted to the engineering system of units in order to be used in the equation for determining Reynolds numbers. Conversion can be made by multiplying centipoises by 0.000672 which gives the absolute viscosity in terms of lb per ft-sec.

Table 2.1 VISCOSITY INDICES FOR VARIOUS MATERIALS

Material	*Temperature, °F*	*S.G. (Approx.)*	*Viscosity* Centipoises	*Viscosity* Lb per Ft-Sec	
Air	32		0.0171	0.0000115	*International Critical Tables*
	70		0.0181	0.0000122	*International Critical Tables*
	212		0.0218	0.0000147	*International Critical Tables*
Water	32	1.000	1.793	0.00121	*International Critical Tables*
	70	0.998	0.984	0.000661	*International Critical Tables*
	120	0.987	0.559	0.000375	*International Critical Tables*
Sucrose, 20% sol.	32	1.086	3.818	0.00256	*International Critical Tables*
	70	1.082	1.916	0.00129	*International Critical Tables*
	176	1.055	0.592	0.000398	*International Critical Tables*
60% sol.	70	1.289	60.2	0.0404	*International Critical Tables*
	176	1.252	5.42	0.00364	*International Critical Tables*
Lub. oil, S.A.E. 10	60	0.9	100	0.0672	*Marks' Handbook*
	150	0.87	10	0.00672	*Marks' Handbook*
S.A.E. 30	60	0.9	400	0.269	*Marks' Handbook*
	150	0.87	27	0.0181	*Marks' Handbook*
Liquid Ammonia	5	0.66	0.25	0.000168	*Refrigeration Data Book*
	80	0.60	0.21	0.000141	*Refrigeration Data Book*
Freon-12	5	1.44	0.33	0.000222	*Refrigeration Data Book*
	80	1.30	0.26	0.000175	*Refrigeration Data Book*
$CaCl_2$ brine, 24% sol.	−10	1.238	12.5	0.00840	*Refrigeration Data Book*
	0	1.234	8.8	0.00591	*Refrigeration Data Book*
	35	1.227	3.7	0.00248	*Refrigeration Data Book*
NaCl brine, 22% sol.	0	1.19	6.1	0.00410	*Refrigeration Data Book*
	35	1.17	2.7	0.00181	*Refrigeration Data Book*
Molasses, heavy dark	70	1.43	6600	4.43	*Gould's Pumps*
	100	1.38	1872	1.26	*Gould's Pumps*
	120	1.31	920	0.618	*Gould's Pumps*
	150	1.16	374	0.251	*Gould's Pumps*
Soybean oil	86	0.92	40.6	0.0273	Eshbach
Olive oil	86	0.92	84.0	0.0565	Eshbach
Cotton-seed oil	60	0.92	91.0	0.061	Eshbach
Milk, whole	32	1.035	4.28	0.00288	Rogers et al.
	68.4	1.03	2.12	0.00143	Rogers et al.
Milk, skim	77	1.04	1.37	0.000922	Bateman and Sharp
Cream, pasteurized, 20% fat	37.4	1.01	6.20	0.00416	Dahlberg and Hening
30% fat	37.4	1.00	13.78	0.00936	Dahlberg and Hening

Viscosity is usually measured by a Saybolt viscometer. The time in seconds is noted for a specified quantity of fluid to flow through a short tube of small bore under prescribed head and temperature condition. The viscosity is reported in seconds.

The Saybolt Universal viscometer is used for fluids of light to medium viscosity. The Saybolt Fural has a tube of larger bore and is used for heavier fluids.

Viscosities in centipoises can be found from Saybolt Universal seconds t by the following equations:

$$\text{Centipoises} = \left(0.226t - \frac{195}{t}\right)\text{S.G.} \tag{2.14}$$

when t varies from 32 to 100 sec

$$= \left(0.220t - \frac{135}{t}\right)\text{S.G.} \tag{2.15}$$

when t is greater than 100 sec

Conversion from Saybolt Fural seconds is made thus:

$$\text{Centipoises} = \left(2.24t - \frac{184}{t}\right)\text{S.G.} \tag{2.16}$$

When t varies from 25 to 40 sec

$$= \left(2.16t - \frac{60}{t}\right)\text{S.G.} \tag{2.17}$$

when t is greater than 40 sec

The critical velocity, that is, the velocity below which streamlined flow exists ($Re = 2130$), is plotted against diameter of pipe for air and water at two temperatures in Fig. 2.5. Note that the

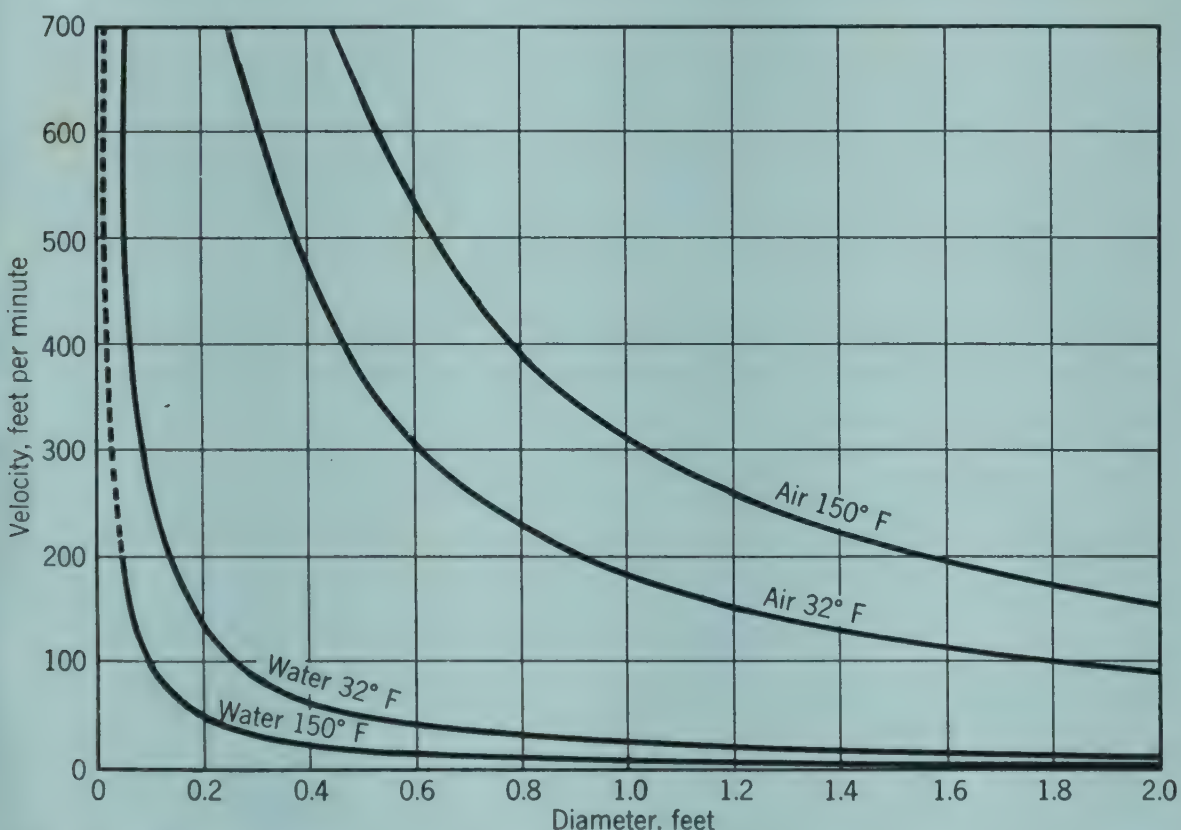


Fig. 2.5. Relationship of velocity and pipe diameter for flow at the critical velocity, $Re = 2130$.

velocity of air increases with temperature but that of water decreases owing to the fact that the viscosity of gases μ increases with temperature but that of liquids decreases. This shows that if turbulent flow is required, heat-exchanger design, e.g., small pipes are not to be desired.

FRICTION LOSSES

The F or friction head loss term in the Bernoulli equation (2.7) represents energy lost or dissipated because of internal fluid resistance, excess turbulence, or resistance of the inner surface of the retainer to flow. Evaluation of this factor involves the Reynolds number, the dimensions of the conduit under consideration, and certain empirical data.

2.12. Darcy's Formula. One of the most widely used formulas for determining the friction loss was developed by Darcy,

$$F = f(l/D)(V^2/2g) \tag{2.18}$$

where l = length of pipe, ft.
D = pipe diameter, ft.
V = linear velocity, ft per sec.
g = acceleration of gravity, 32.2 ft per sec.
f = coefficient, dimensionless.

The coefficient f is closely related to Reynolds number Re, but the relationship is not definite enough for general use in a mathematical form.

2.13. Friction-Factor Diagram. Factor f in the Darcy formula (2.18) is some function of Reynolds number and the degree of roughness of the inside surface of the conduit. Moody[19] has related these factors as shown in Table 2.2 and Fig. 2.6. The

Table 2.2 ROUGHNESS INDICES FOR VARIOUS TYPES OF PIPE

Pipe Material	*Roughness Factor* ϵ
Riveted steel	0.003 –0.03
Concrete	0.001 –0.01
Wood stave	0.0006–0.003
Cast iron	0.00085
Galvanized iron	0.0005
Asphalted cast iron	0.0004
Commercial steel or wrought iron	0.00015
Drawn tubing	0.000,005

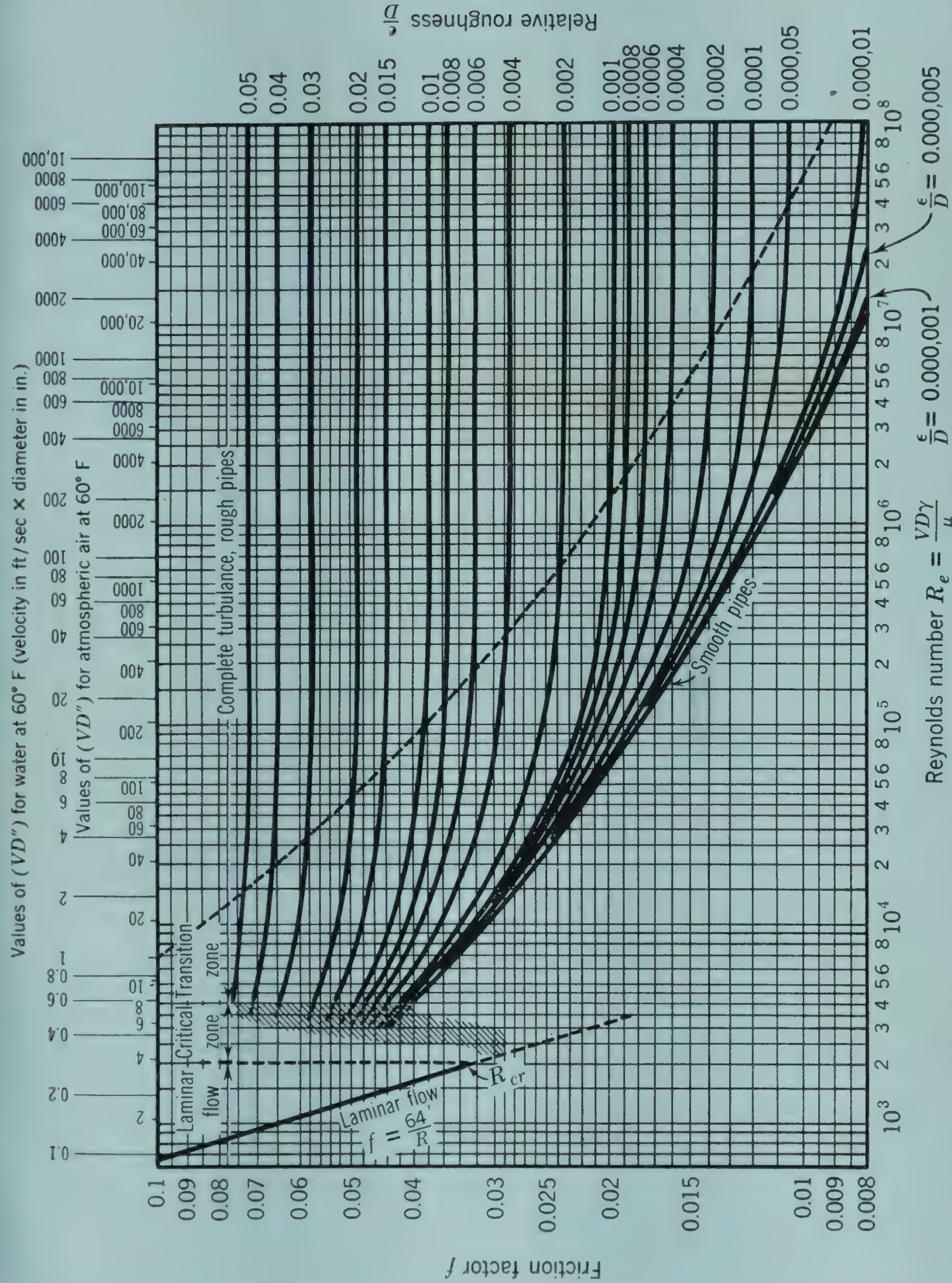


Fig. 2.6. Reynolds number vs. the friction factor f for pipe of various relative roughness.

relative roughness factor is the roughness factor ϵ divided by the pipe diameter in feet. The relative roughness factor for a particular pipe is referred to in Fig. 2.6 and identifies the curve to be used for selecting a satisfactory f value.

For example, a 3-in. (inside diameter) commercial steel pipe has a relative roughness of 0.0006. This value identifies the proper curve to be used in Fig. 2.6. If Reynolds number is found to be 9×10^4, the friction factor f is 0.021. Friction factors for water and atmospheric air can be determined from the values of VD from the top of Fig. 2.6. Air is flowing at 900 ft per min in a 20-in. galvanized iron pipe. The relative roughness is 0.003, which identifies the proper curve of Fig. 2.6. The product of VD is 300, which provides a friction factor f of 0.027. If Reynolds number is desired, it can be read directly from the VD position. It is 1.4×10^5 in this case.

Note that the velocity term V shows in the formula for determining Reynolds number (2.8), in Darcy's formula (2.18), and in the Bernoulli formula (2.7). Solution of a problem in which the velocity is known is a straightforward arithmetical procedure. On the other hand, if velocity is to be determined, solution must be by trial and error or by a graphical procedure. Trial-and-error solutions are usually satisfactory, but the graphical method gives more accurate results. This method is demonstrated by the following example.

Example. How many gallons of water per minute will flow through 150 ft of 2-in. pipe under a 15-ft head?

The Bernoulli factors that apply are,

$$h_1 - F = V^2/2g$$

and since

$$F = f(l/D)(V^2/2g)$$

$$h_1 - f(l/D)(V^2/2g) = V^2/2g$$

Substituting known values and solving,

$$15 - f\frac{150}{\frac{2}{12}}\frac{V^2}{64.4} = \frac{V^2}{64.4} \qquad \text{and} \qquad 15 - 14fV^2 = \frac{V^2}{64.4}$$

Transpose and divide to place f and V^2 on opposite sides of the equal sign and equate both to a variable thus:

$$14f = (15/V^2) - (1/64.4) = C$$

Plot the two equations

$$14f = C \qquad \text{and} \qquad (15/V^2) - (1/64.4) = C$$

for a number of values of V. The point at which the curves intersect is the solution (see Fig. 2.7). This procedure is known as Newton's method of solution and can be used to solve many algebraic equations which cannot be easily solved by other methods.

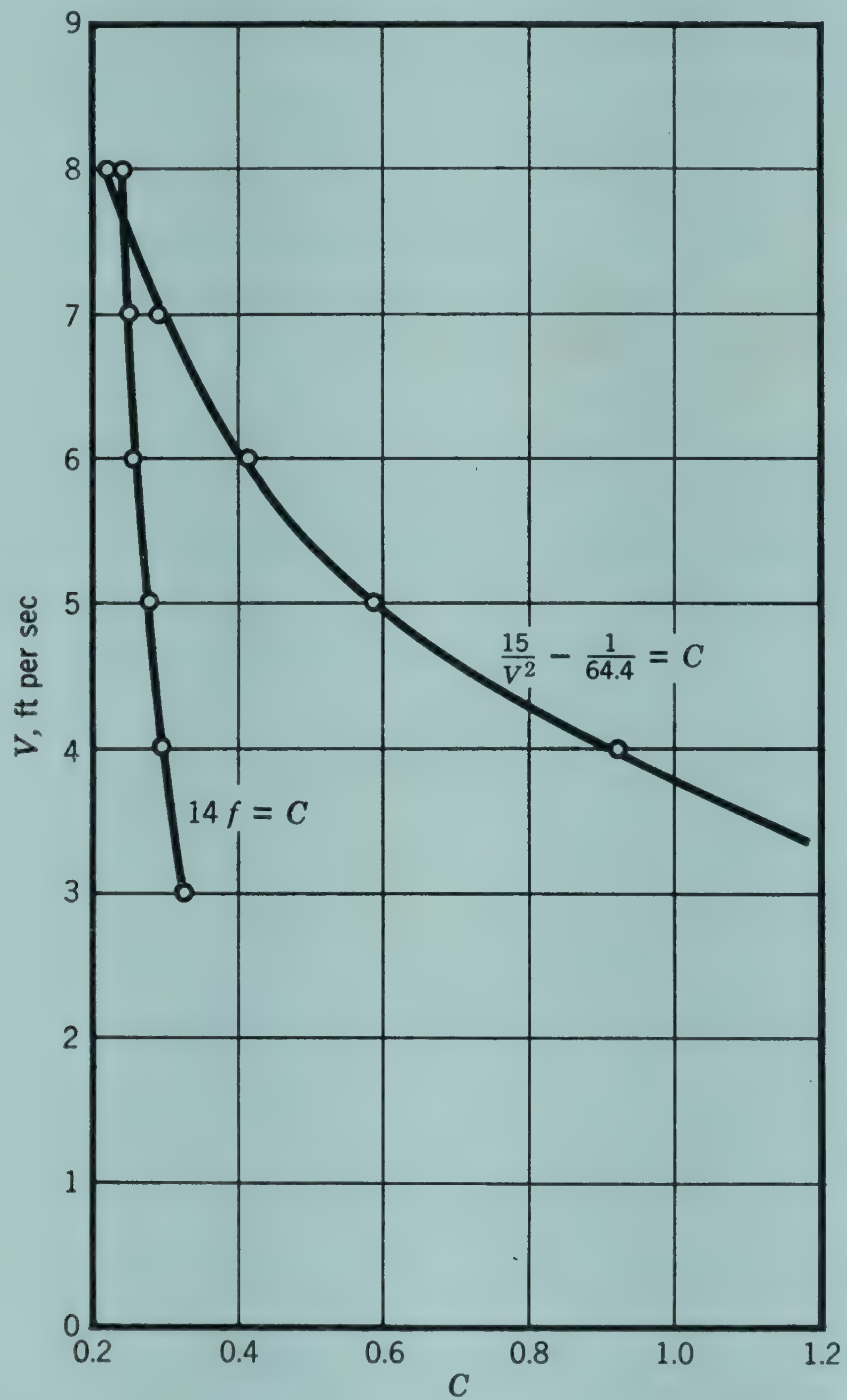


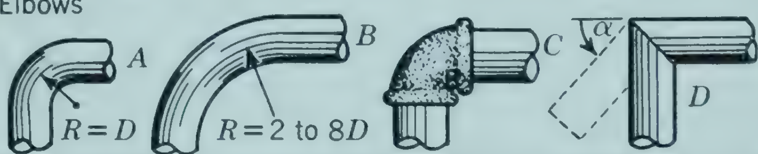
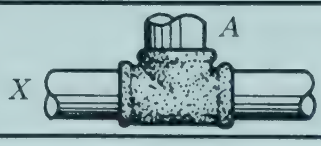
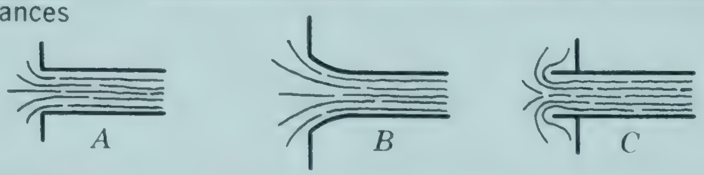
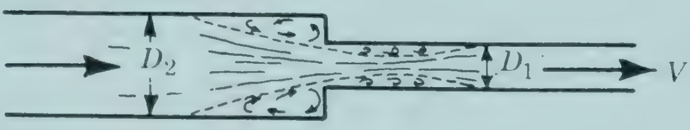
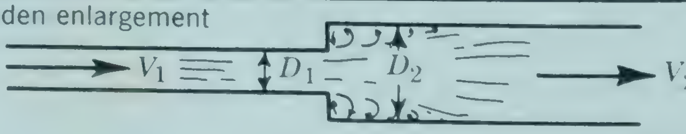
Fig. 2.7. Graphical solution of velocity problem.

2.14. Resistance of Fittings. Pipe and conduit fittings, because of restrictions to flow, sharp projections, abrupt change in

shape and dimensions, etc., may cause a significant loss of energy which further adds to the F factor of the Bernoulli equation. The characteristics of this loss have not been sufficiently rationalized for mathematical treatment. Considerable empirical data are available and, although incomplete as regards many fittings and fluids, are sufficiently accurate for most design work.

The data are usually presented in one of two ways, either as loss in pressure head as a decimal of $V^2/2g$ or as an equivalent length of pipe. Resistance data for a number of fittings expressed as a fraction of the velocity are tabulated in Table 2.3. The re-

Table 2.3 FRICTION LOSS FACTORS K

Nature of resistance		K
Valves, fully open	Gate globe angle	0.15 7.5 4.0
Elbows 		A, 0.50 B, 0.25 C, 1.50 D, $1.25\left(\frac{\alpha}{90}\right)^2$
Tees 		XA, 1.50 XB, 0.50
Discharge nozzles 	Plain / Ring / Spray	A, 0.01 – 0.03 B, 0.01 – 0.04* C*
Entrances 		A, 0.50 B, 0.05 C, 1.00
Sudden contraction 		D_1/D_2: 0.1, 0.3, 0.5, 0.7, 0.9 K: 0.362, 0.308, 0.221, 0.105, 0.015
Sudden enlargement 		$\frac{(V_1 - V_2)^2}{2g}$

* Varies, use manufacturers values.

sistance expressed as an equivalent length of straight pipe in terms of pipe diameter is $40K$. For example, a common elbow C with a K of 1.50 would have the same resistance or produce the same pressure drop as a length of connecting pipe equal to 1.5 times 40 or 60 diameters. It is frequently convenient to use this equality in determining pressure losses in lines that include various fittings.

2.15. Energy Losses Due to Sudden Velocity Changes. When a fluid flowing in a pipe is forced to change velocity, a certain amount of energy is lost as heat energy because of turbulence and work energy due to localized velocity variation. A list of conditions in which this combined effect is important and methods used for its determination are given in Table 2.3. Except for the enlargement condition, the loss factor K has been determined experimentally.

The loss factors for sudden contraction, sharp-edged entrances, and nozzles involve two phenomena: turbulence, which has been discussed, and stream contraction. Because of inertia, an element of fluid does not necessarily follow the walls of the retaining structure. For example, the stream of water after leaving the point of sudden contraction in Table 2.3 is smaller in diameter than the pipe and has a velocity higher than it has farther along in the pipe. The point at which this stream diameter is a minimum is called the *vena-contracta*. Vigorous turbulence in the region of the vena-contracta between the wall of the pipe and the flowing stream, i.e., when there are no reverse eddies, causes considerable energy loss. A rounded approach avoids the formation of a vena-contracta, and a cone of expansion with a slope angle of 7 degrees or less permits a change in velocity with a minimum energy loss. For a detailed study of these losses, the reader should refer to a textbook on fluid mechanics or to detailed specialized reports.

2.16. Pressure Drop in Heat Exchangers. The resistance to air flow or pressure drop through heat exchangers is important in such installations as refrigeration plants, air conditioning units, driers, etc. In general, it is expedient to use the pressure-drop data supplied by the manufacturer for the exchanger in question. A general method of calculating this effect for certain conditions will be found useful and follows.

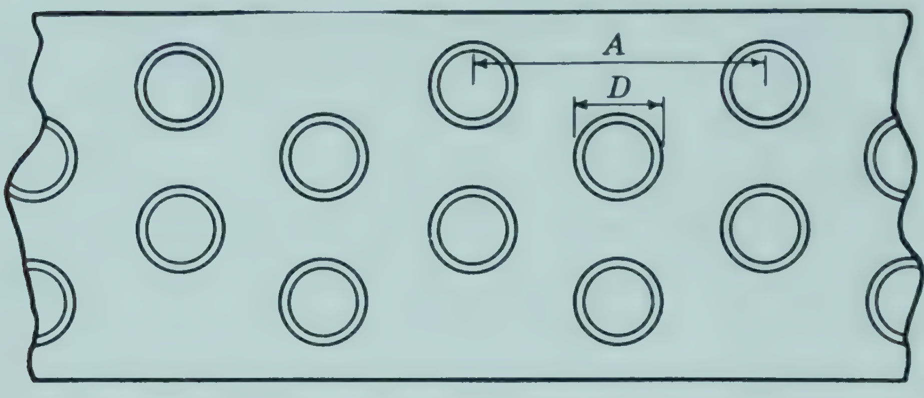


Fig. 2.8. Cross section of a heat exchanger.

If the exchanger is made of a series of parallel tubes, the pressure drop through it can be calculated by the following equations, which are the results of studies by a number of investigators.[20] The formulas which should be referred to Fig. 2.8 follow.

$$p = \frac{4fE\gamma V^2}{2g} \tag{2.19}$$

$$f = 0.75\left(\frac{CV\gamma}{\mu}\right)^{-0.2} \tag{2.20}$$

where p = pressure drop, lb per sq ft.
E = number of rows of tubes normal to fluid stream.
γ = fluid specific weight, lb per cu ft.
V = maximum velocity through the minimum cross section, ft per sec.
C = clearance between tubes in a row, $A-D$ in Fig. 2.8, ft.
μ = viscosity, lb per ft-sec.

Equation 2.19 is probably reliable to within 25 per cent for pitch distances A of 1.25 to 1.50 tube diameters, which is the normal commercial spacing. Flow is probably turbulent if $(A-D)V\gamma/\mu$ is greater than 40.

Baffled or finned exchangers require an involved method of calculation and will not be discussed here.

2.17. Pressure Drop through Agricultural Products. Ventilating, drying, and dehydrating of agricultural products usually involve forcing air through a mass of the product. The relationship of the rate of flow through the mass to the depth of material or distance of air travel and the pressure drop through the material is important since the power requirement, fan or blower

selection, and drying characteristics are directly related to these. The calculation of fan or blower requirements and drying characteristics are treated in Chaps. 5 and 11, respectively.

Considerable work has been done on the characteristics of fluid flow through soils and through granular and other material related to chemical engineering. Unfortunately, these procedures have not been verified for application to agricultural products. Although some work has been done toward rationalization of the resistance relationships, current usable data are mostly empirical in nature.

The resistance of a material to air flow is some function of the surface characteristics and the size and shape of the voids. Consider the variations in these factors if we attempt to compare such agricultural commodities as flax seed, ear corn, walnuts, beet seed, oats, and hay. These factors plus natural biological variation due to moisture content, varieties, seasons, and geography thus far have complicated the complete rationalization of resistance data.

Chilton and Colburn (*Ind. and Engr. Chem.* 23:913–919. 1931) correlated the available resistance data for many uniform granular solid particles used in chemical-engineering porous beds by means of a modified Reynolds number:

$$Re_m = \frac{D_p V_0 \gamma}{60\mu}$$

where D_p = nominal particle diameter, ft.
V_0 = air velocity, cu ft per min sq ft.
γ = fluid specific weight, lb per cu ft.
μ = viscosity, lb per ft sec.

The modified Reynolds number was plotted against the friction factor f, in a manner comparable to Fig. 2.6. Although there was no distinct break between turbulent and laminar flow, turbulent flow seemed to persist above an Re_m of 100 and a laminar flow below 20. A single break point could be indicated at $Re_m = 40$. The analysis also showed the following relationship for pressure drop through the material:

For laminar flow:

$$p' = K_1 \frac{\mu L V_0 A_f}{D_p^2}$$

For turbulent flow:

$$p' = K_2 \frac{\mu^{0.15} L^{0.85} V_0^{1.85} A_f}{D_p^{1.15}}$$

where p' = pressure drop through the mass, in. water.
L = mass depth, ft.
K_1, K_2 = proportionality constants.
A_f = wall effect factor, dimensionless (this factor will equal one for most agricultural installations).

This treatment of itself cannot determine resistance data for a particular agricultural material. It can be used, however, for evaluating observed data and determining the limits to which observed data can be extrapolated.

Resistance data for a number of agricultural products have been observed by various investigators by noting the relationship between material depth, static pressure, and air flow rate. Data for a few products * are listed in Table 2.4. These data can

Table 2.4 RATE OF AIR FLOW (CU FT PER MIN SQ FT) FOR VARIOUS AIR PRESSURES

Depth of Grain, ft	*Air pressure, in. of water*					
	0.10	0.25	0.50	1.00	2.00	3.00
			Wheat			
0.5	6.1			54.0	80.0	92.0
1.0	3.8	9.8	18.5	31.5	49.5	62.0
2.0	2.1	4.7	9.2	18.5	31.8	41.5
4.0	1.2	2.1	6.5	11.5	20.5	27.5
8.0	0.7	1.3	2.8	6.9	13.0	18.5
			Shelled Corn			
0.5	21.0	35.0	51.5	76.0	120.0	
1.0	14.0	24.0	36.0	59.0	80.0	102.0
2.0	10.3	17.3	25.8	38.3	57.0	72.0
4.0	5.9	10.4	16.0	24.3	37.5	48.0
8.0	3.2	5.8	9.3	14.7	23.2	30.4
			Soybeans			
0.5	26.2	44.6	67.0	100.0		
1.0	17.7	33.5	49.0	76.0	115.0	
2.0	12.4	22.0	33.0	49.6	74.5	95.0
4.0	6.6	12.1	19.0	30.0	47.0	62.0
8.0	4.3	8.2	12.9	20.7	33.0	43.5

probably be used for design with satisfactory results, but some variations are to be expected as indicated above.

Resistance data for hay, although exhibiting the same performance characteristics as grain, are too variable for specific

* For additional data see Stahl,[27] Engineering Data on Grain Storage.

recommendation. Schaller et al.[23] recommend that, for hay drying, a total pressure drop of 0.75 in. of water be used in selecting a fan for hay 15 ft deep. For depths of 6 to 8 ft of hay a pressure drop of 0.5 to 0.6 in. may be used. The rate of air flow should not be less than 10 cu ft per sq ft per min. A higher rate is preferable.

An acceptable mathematical relationship of the variables is:

$$p' = KV_0{}^mL^n \tag{2.21}$$

where V_0 = rate of air flow, cu ft of air at atmospheric pressure, and temperature per sq ft of floor area per min. Note that the linear rate through the mass would be V_0 divided by the porosity of the mass.

K = a constant that depends upon the characteristics of the material.

p' = pressure drop through the mass, in. of water.

m = an exponent that varies from material to material and varies somewhat with depth for any one material. Observed values vary from 1.1 to 2.0 approximately with a value of about 1.5 being an indicated mean.

L = depth of material, or distance of air movement through mass, ft.

n = an exponent that varies from 1.0 to 1.1, approximately.

Note that the pressure drop (head loss) is indicated in inches of water. To convert inches of water to pounds per square inch multiply by 0.0362.

If n and m were 1.0 and 2.0 respectively, equation 2.21 would become

$$p' = KV_0{}^2L \tag{2.22}$$

which is essentially the Darcy friction formula. Now if the exponent of V_0 were 1, flow would be streamlined. Since the actual observed exponents or values of n and m are such that the exponent of V_0 in equation 2.21 is between 1 and 2, the flow is probably a combination of streamlined and turbulent. Furthermore, if this were the case, material with small void spaces would have m values approaching 1.0. A review of the literature shows a tendency in this direction.

2.18. Pressure Drop Through Floors. The perforated floor or wall which retains an agricultural product being dried offers resistance to air flow in addition to the resistance of the material. Henderson [10] found the following experimental relationship for perforated floors.

$$V_0 = 3000(\text{p.ct})p^{0.52} \tag{2.23}$$

or approximately

$$p = \frac{10^{-6}}{9}\left(\frac{V_0}{\text{p.ct}}\right)^2 \tag{2.24}$$

where p = pressure drop, in. water
V_0 = rate of flow, cu ft per min sq ft
p.ct = per cent of opening, expressed as a decimal.

When material is placed on a perforated floor, the effective amount of floor opening is decreased. Theoretically, we would expect the effective area to be reduced to an amount equal to the percentage void space in the material. A test by Henderson [10] using shelled corn with 40 per cent voids confirms this expectation. Consequently, if we assume this condition to hold for all material, the expression for pressure drop is

$$p' = \frac{10^{-6}}{9}\left(\frac{V_0}{\text{p.ct}\ v}\right)^2 \tag{2.25}$$

in which v is the amount of void space in the material expressed as a decimal.

2.19. Branching System Design. Frequently a system of conduits must be designed so that the flowing fluid is divided in some proportion among a number of branching lines. An air conditioning or ventilating system serving a number of locations is an example.

Where a dividing system of conduits is to be designed the *equal-pressure-drop* method is the most usable. This method is illustrated by the following example.

Example. A seed cleaning house is to install a hood over each machine to exhaust dust arising from the cleaning operation. A schematic plan of the system is shown in Fig. 2.9. The steps to follow are:

1. Determine the pressure drop between the system outlet and the hood at the greatest distance from the outlet. The optimum velocity in the pipe will control the pressure. In this case, 1000 ft per min is considered desir-

able. The dust and dirt which will be removed is finely divided and will remain suspended at this velocity.

Hood D is the greatest distance from the datum R. Consider a single 8-in. pipe between these points since the velocity will be approximately 1000 ft per min to deliver 300 cu ft per min. The single elbow resistance may be considered equivalent to that of a length of pipe of 50 diameter or 33 ft, and the hood 20 diameters or 17 ft. The total effective length is 250 ft. The pressure drop between D and R determined by the previous procedure is 0.41 in. of water and is assumed to be uniform through this distance.

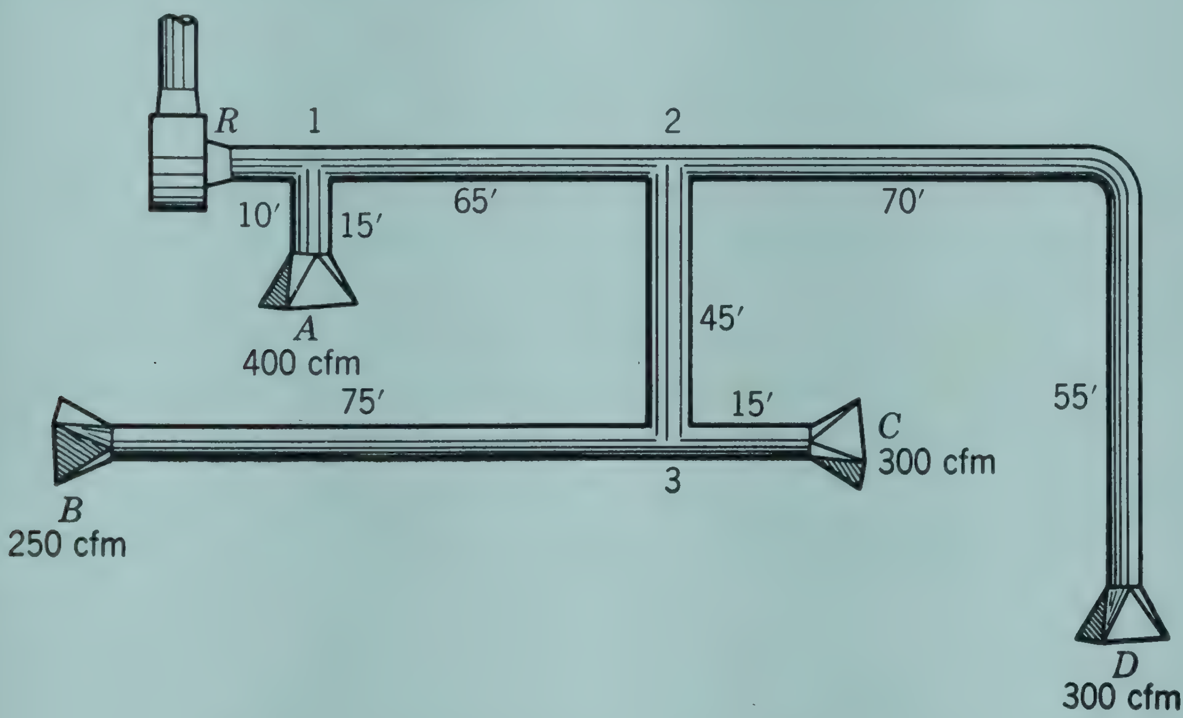


Fig. 2.9. Schematic drawing of an exhaust system.

2. Adjust the size of the trunk and laterals so that the pressure drop between each hood and the datum R is the same as in the longest line, 0.41 in. water. The drop in trunk 1–2 is $^{65}\!/_{250} \times 0.41$ or 0.107 in. of water. To handle the 850 cfm, the diameter here must be 11 in. The drop in trunk R–1 is $^{10}\!/_{250} \times 0.41$ or 0.0164 in., and, to handle 1250 cfm, the diameter must be 13 in.

The pressure drop in line R–2 is $(10 + 65)/250$ of 0.41 or 0.123 in. Therefore, the drop in line 2–C and 2–B must be $0.41 - 0.123$ or 0.287 in.

Line 2–C is 60 ft long and contains two elbows and a hood. The total elbow and hood resistance is 120 pipe diameters. Trial-and-error procedure will show that a 7½-in. pipe of equivalent length of 135 ft will give a pressure drop of nearly 0.287 in. when delivering 300 cu ft per min.

The pressure drop in section 2–3 which includes the elbow at 2 would be $\dfrac{45 + 50\,{}^{7.5}\!/_{12}}{60 + 120\,{}^{7.5}\!/_{12}}$ of 0.287 or 0.218 in. Therefore the drop in line B–3 must be 0.287–0.218 or 0.059 in. Line B–3 resistively is 75 + 70 diameter in length. The same trial-and-error procedure will show that a 9½-in. pipe is necessary.

The pressure drop in R–1 is $^{10}/_{250}$ of 0.41 or 0.016 in. Therefore, the drop in 1–A is 0.41 – 0.016 or 0.394 in. The effective length of 1–A is 15 + 70 diameter. Under these conditions and 400 cu ft per min, a 4-in. pipe is adequate.

Line 2–3 is 45 ft long. The pressure drop is 0.218 in., and it carries 550 cu ft per min. The diameter must be 8 in.

It is advisable to provide dampers or provision for them if needed to balance the system. Certain values, particularly hood and elbow friction, are subject to variation which may require adjustments after installation.

2.20. Compressibility Error. Air that is subjected to a pressure to force it through a series of pipes, a mass of grain, or a heat exchanger is compressed so that γ_1 is not equal to γ_2. In most calculations where drying and ventilation problems are being considered, air is assumed to be incompressible to simplify calculations, atmospheric pressure being used throughout. Pressures under these conditions seldom exceed 10 in. of water and are usually in the order of 4 or less. The error resulting by neglecting compression is dependent upon the absolute pressures. It would be only 2.5 per cent if 10 in. of water were the operating pressure.

2.21. Optimum Rates of Flow. The question frequently arises as to whether one should have a small pipe or conduit with high velocity or a large pipe with low velocity. Although each installation should be analyzed carefully from the standpoint of initial cost, power requirement, noise level, and operating costs, the following general suggestions can be used as a guide.

Velocities of 4 to 6 ft per sec are usually best for water. Ten feet per second may be used if the system resistance is low. Where noise is not a problem, air systems may be designed for velocities of 1000 to 1500 ft per min. Velocities up to 2000 ft per min may be used in large pipes.

The reader should realize that these values are general and that frequently values above or below these should or could be used.

FLOW OF GRANULAR MATERIALS

Grain, ground feed, and other similar materials flow in an entirely different manner than liquids.

2.22. Rate of Flow. Ketchum[15] found that the rate of flow of wheat from an orifice is independent of the head and varies as the cube of the orifice diameter. This phenomenon can be ex-

plained in this way: as soon as flow starts, the grain tends to form a bridge above the orifice. Grain falling from the dome of the bridge region is replaced by grain from above the dome, the grain above the orifice being discharged first.

2.23. Angle of Repose. When a granular material is permitted to flow from a point into a pile as shown in Fig. 2.10, the shape of the pile is characteristic of the material. The angle ϕ which the side of the pile makes with a horizontal is called the angle of repose. For any material, it varies with the moisture

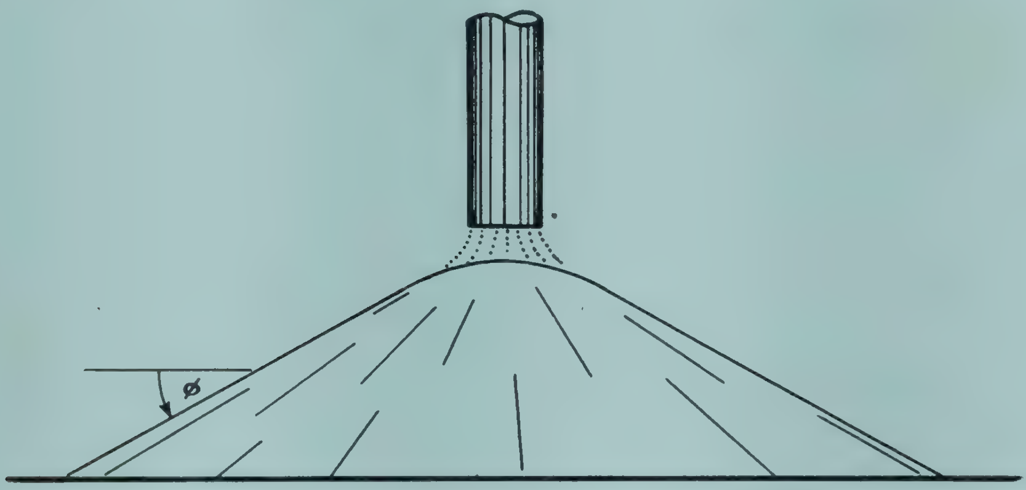


Fig. 2.10. Angle of repose of grain.

content and amount of foreign material present, increasing with an increase in either. The tangent of this angle is recognized as the coefficient of friction of the material on itself.

This property is important in material transfer since it affects the capacity of belt conveyors and other bulk transfer devices and partially determines the minimum slope of floors in self-emptying bins, coefficient of friction of grain on the bin material being another factor. Some materials, particularly those that have been produced by grinding, have such steep repose angles that they are not completely self-flowing. Agitation is usually necessary to maintain flow.

2.24. Coefficient of Friction. Granular materials will not flow through pipes or chutes unless the pitch is sufficient to overcome the coefficient of friction of the material upon the conduit. This characteristic determines the minimum pitch of conduit intended to move materials by gravity. Grain or other granular materials will flow in a conduit at a flatter angle if it is moving when introduced into the conduit. If a system is designed on this

basis, trouble may arise from accidental stoppage since starting flow may be difficult if a minimum pitch is used.

Coefficients of a few grains as reported by Ketchum[15] are listed in Table 2.5.

Table 2.5. COEFFICIENTS OF FRICTION OF VARIOUS KINDS OF GRAIN ON BIN WALLS

	Weight of a Cubic Foot Loosely Filled into Measure, lb	*Coefficients of Friction*				
		Grain on Grain	*Grain on Rough Board*	*Grain on Smooth Board*	*Grain on Iron*	*Grain on Cement*
Wheat	49	0.466	0.412	0.361	0.414	0.444
Barley	39	0.507	0.424	0.325	0.376	0.452
Oats	28	0.532	0.450	0.369	0.412	0.466
Corn	44	0.521	0.344	0.308	0.374	0.423
Beans	46	0.616	0.435	0.322	0.366	0.442
Peas	50	0.472	0.287	0.268	0.263	0.296
Tares	49	0.554	0.424	0.359	0.364	0.394
Flax seed	41	0.456	0.407	0.308	0.339	0.414

REFERENCES

1. Barnard, W. N., et al. *Elements of Heat Power Engineering,* Part III. Wiley. 1933.
2. Bateman, G. M., and Sharp, P. F. A Study of the Apparent Viscosity of Milk as Influenced by Some Physical Factors. *J. Agr. Research* 36:647–674. 1928.
3. Bruhn, H. D. Flow of Air Through Chopped Hay in Mow Drier. *Agr. Eng.* 28:202–204, 207. 1947.
4. Crane Co. *Engineering Data on Flow of Fluids in Pipes and Heat Transmission.* Chicago. 1935.
5. Dahlberg, A. C., and Hening, J. C. Viscosity, Surface Tension, and Whipping Properties of Milk and Cream. *N. Y. Expt. Sta. Tech. Bull.* 113. 1925.
6. Davis, Roy B., Jr. Mow-Drying Chopped and Long Alfalfa Hay. *Agr. Eng.* 28:105–108. 1947.
7. Dodge, R. A., and Thompson, M. J. *Fluid Mechanics.* McGraw-Hill. 1937.
8. Guillou, René. Forced Air Flow in Drying Hay. *Agr. Eng.* 27:519–520. 1946.
9. *Heating, Ventilating, and Air Conditioning Guide.* American Society of Heating and Ventilating Engineers. 1943.

10. Henderson, S. M. Resistance of Shelled Corn and Bin Walls to Air Flow. *Agr. Eng.* 24:367–369, 374. 1943.

11. Henderson, S. M. Resistance of Soybeans and Oats in Storage to Air Flow. *Agr. Eng.* 25:127–218. 1944.

12. Hendrix, A. T. Air Flow Through Baled Hay. *Agr. Eng.* 26:369–371. 1945.

13. Hendrix, A. T. Observations on the Resistance of Hay to Air Flow. *Agr. Eng.* 27:209–212. 1946.

14. Hendrix, A. T. Resistance of Hay to Air Flow. *Agr. Eng.* 26:369–371. 1945.

15. Ketchum, M. S. *The Design of Walls, Bins and Grain Elevators.* Third Edition. McGraw-Hill. 1919.

16. Kratz, A. P., and Fellows, J. R. Pressure Losses Resulting from Changes in Cross-Sectional Area in Air Ducts. *Ill. Eng. Expt. Sta. Bull.* 300. 1938.

17. Lansford, W. M. Loss of Head in Flow of Fluids Through Various Types of One- and One-Half-Inch Valves. *Ill. Eng. Expt. Sta. Bull.* 340. 1943.

18. Marks, L. S. *Mechanical Engineers Handbook.* McGraw-Hill. 1941.

19. Moody, L. F. Friction Factors for Pipe Flow. *Trans. A.S.M.E.* 66:671–684. 1944.

20. Perry, J. H. *Chemical Engineers' Handbook.* Third Edition. McGraw-Hill. 1950.

21. Rogers, L. A. *Fundamentals of Dairy Science.* Second Edition. Reinhold. 1935.

22. Rouse, Hunter. *Elementary Mechanics of Fluids.* Wiley. 1946.

23. Schaller, J. A., Mitchell, N., Dickerson, W. H., Jr. *Barn Driers; Principles of Design, Installation, and Operation.* Tennessee Valley Authority. 1945.

24. Shedd, Claude K. Resistance of Hay to Air Flow and Its Relation to Design of Barn Hay-Curing Equipment. *Agr. Eng.* 27:169–170. 1946.

25. Shedd, Claude K. Resistance of Ear Corn to Air Flow. *Agr. Eng.* 26: 19–20, 23. 1945.

26. Spaugh, O. H. Air Flow Through Beds of Dehydrated Vegetables. *Food Tech.* 2:33–38. 1948.

27. Stahl, B. M. *Engineering Data on Grain Storage. Agr. Eng. Data 1.* American Society of Agricultural Engineers. 1948.

28. Stirniman, E. J., Bodnar, G. P., and Bates, E. N. Tests on Resistance to the Passage of Air Through Rough Rice in a Deep Bin. *Agr. Eng.* 12:145–148. 1931.

29. *The Refrigeration Data Book.* The American Society of Refrigeration Engineers. 1936.

30. Vennard, J. K. *Elementary Fluid Mechanics.* Second Edition. Wiley. 1947.

31. Weaver, John W., Jr., Grinnells, C. D., and Louvern, R. L. Drying Baled Hay with Forced Air. *Agr. Eng.* 28:301–304, 307. 1947.

PROBLEMS

1. Find Reynolds number for milk at 70°F flowing at 20 gal per min in sanitary tubing with 1⅜-in. inside diameter. Milk weighs 64.2 lb per cu ft. What would be the diameter of a tube in which streamlined flow could be expected?
2. A tank 3½ ft in diameter (D) contains 5 ft of water and is fitted at the bottom with a ¾-in. globe valve. What is the initial rate of discharge? How long will it take to empty the tank if the valve is completely open? Note that

$$dQ = A\sqrt{\frac{2gh}{K+1}}\,dt \quad \text{and that} \quad t = \frac{\pi D^2}{4A}\sqrt{\frac{K+1}{2g}}\,\frac{dh}{\sqrt{h}}$$

   How long will it take if a gate valve is used?
3. Milk is to be lifted 12 ft through 30 ft of sanitary pipe that contains 2 elbows. Assuming a pump efficiency of 80 per cent, how much power will be required to pump at a rate of 60 gal per min if ¾-in. pipe is used? If 1½-in. pipe is used?
4. How much power would be required to pump molasses at 70°F (S.G., 1.43) through the system of problem 3 at a rate of 1½ gal per min, assuming a pump efficiency of 70 per cent?
5. Fifteen cubic feet of air per minute per square foot of floor are to be moved vertically through a crib of shelled corn 5 ft deep. The area of the floor is 120 sq ft, and the connecting pipe is 12 in. in diameter and 35 ft long. What is the power requirement, assuming fan efficiency to be 75 per cent? If the diameter of the connecting pipe is increased to 18 in., how much power will be required?
6. Air is flowing through a conduit system at 1400 cu ft per min. An 8-in. galvanized iron pipe enlarges abruptly to 16 in. The 16-in. section is 20 ft long. It decreases abruptly at the end of the section to 8 in. in diameter. Would the horsepower requirement increase or decrease, and by how much, if the central section were reduced in diameter to 8 in.?
7. For a specific fluid, it is convenient to have the friction loss available in terms of the rate of flow, say in gallons per minute, and the diameter in inches. For smooth tubes, in the range of Re from 5000 to 100,000, the friction factor f, is given by the Blasius equation,

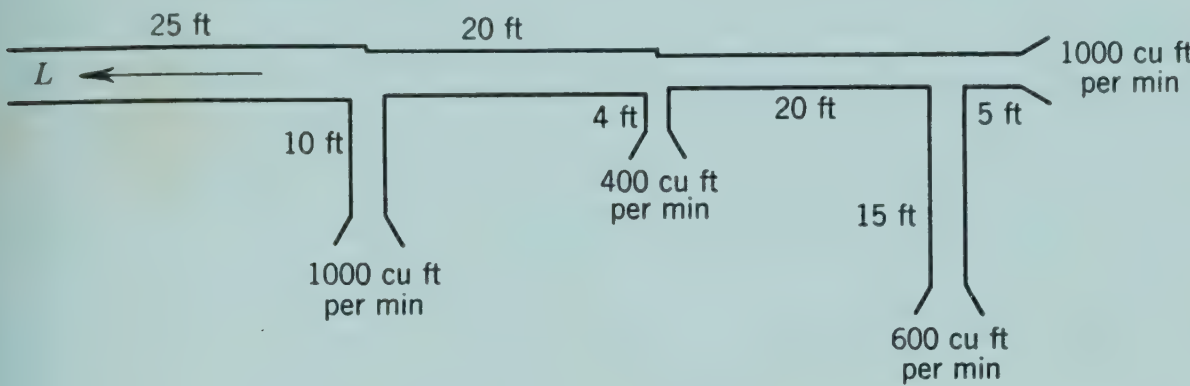
$$f = 0.316/Re^{0.25}$$

Find the friction constant c in

$$F/1 = \frac{c(\text{gal per min})^{1.75}}{(D')^{4.75}}$$

for a fluid of viscosity of 20 centipoises and a density of 70 pounds per cu ft in smooth tubes of inside diameter D' in.

8. Determine the size of the pipes required for the installation in the accompanying figure.



9. A milk heater with 6 passes of 1½-in. sanitary tubing (1.38 in. ID) is 10 ft long set up for testing. When 20 gal per min are being delivered on a cold test at 50°F, the height of milk in a piezometer at the inlet is 8.4 ft. Find the friction loss in a return bend in equivalent velocity heads; in equivalent pipe diameters.

CHAPTER 3

Fluid-Flow Measurements

NOMENCLATURE

A = cross-sectional area, sq ft.
b = gage elevation, ft.
C = a proportionality constant.
c_p = specific heat at constant pressure, Btu per lb °F.
Δc, Δd = pressure head, in. of mercury.
D = diameter, in.
ΔD = pressure head, ft of mercury.
d = diameter, in.
H = head, i.e., energy in ft-lb per lb of fluid.
h = height, ft.
i = electrical current, amp.
K = thermometer factor, dimensionless.
k = a proportionality constant.
m = a ratio.
n = an area ratio, dimensionless.
p_g = percentage, a decimal.
p = pressure, lb per sq in.
p_i = pressure, in. of water.
Q = a quantity rate.
R = electrical resistance, ohms.
t = temperature, °F.
t_c = 36.5 degrees minus room temperature, °C.
V = velocity, ft per unit of time.
w = weight, lb.
W_g = gas rate, lb per min.
γ = specific weight, lb per cu ft.
θ = time, sec.

PRESSURE AND VELOCITY MEASUREMENTS

Accurate and practical methods must be used by the processing engineer for determining the pressure and velocity of fluids under study. Considerable knowledge about this phase of fluid mechanics is important for the researcher, the designer, the contrac-

tor, the operator, and the trouble shooter. Types of equipment and techniques are diverse, therefore satisfactory selection and performance can be assured only if the individual is well versed in the various types of equipment and methods of procedure.

Various considerations must be given the selection, installation, and use of the equipment for a specific case. Mobility, accuracy, constancy of calibration, sensitivity, range of operation, ruggedness, reliability, and longevity all must be considered. In the following sections these points will be clarified relative to the pressure and flow measurement problems that might confront the processing engineer.

3.1. Pressure Observations. The pressure heads represented by the Bernoulli equation are in terms of a column of the fluid under consideration having a height expressed in feet. In practice, pressures are usually indicated in pounds per square inch, inches of mercury, or inches of water. Pounds per square inch are used for relatively high pressures, inches of water for low pressures, and inches of mercury for pressures less than atmospheric or vacuum. The pressure heads $H = 144p/\gamma$ in the Bernoulli equation (2.7) are in terms of the ft-lb of energy per pound of fluid, equivalent to the energy of a column h feet high and

$$H = h \tag{3.1}$$

The pressure per square inch is expressed by the following equation:

$$p = \frac{h\gamma}{144} \tag{3.2}$$

where p = pressure, lb per sq in.
h = pressure head of fluid, ft.
γ = specific weight of fluid, lb per cu ft.

The pressure in inches of water is derived and expressed thus:

$$p_{\text{in. water}} = \frac{12h\gamma}{62.4} \tag{3.3}$$

Similarily, the expression in terms of inches of mercury is

$$p_{\text{in. mercury}} = \frac{12h\gamma}{847} \tag{3.4}$$

By solving equations 3.2, 3.3, and 3.4 simultaneously, the relationship between pressures in pounds per square inch, inches of water, and inches of mercury can be found. This relationship or the factors used for converting from one system to another are shown in Table 3.1.

Table 3.1 PRESSURE CONVERSION FACTORS

Multiply	*by*	*to Obtain*
Lb per sq in.	27.684	In. of water
Lb per sq in.	2.036	In. of mercury
In. of water	0.0361	Lb per sq in.
In. of water	0.0736	In. of mercury
In. of mercury	0.491	Lb per sq in.
In. of mercury	13.6	In. of water

Pressures less than atmospheric, usually referred to as vacuum, are usually expressed in inches of mercury. Pressures just above atmospheric such as are encountered in ventilation, air conditioning, and drying, are in inches of water. Higher pressures are expressed in pounds per square inch. In testing and research the system which best fits the needs, equipment on hand, etc., is used, although the system adaptation usually follows as indicated.

Pressures are referred to as "gage" or "absolute." Gage pressures indicate the pressure above atmospheric. Absolute pressure is gage pressure plus atmospheric. Atmospheric is assumed standard at 14.7 lb per sq in. unless otherwise indicated.

STATIC PRESSURES

3.2. General Considerations. Pressures are referred to as static or dynamic. Static pressures are those resulting from pressure and elevation and indicate forces perpendicular to the walls of the container. Dynamic pressures which result from the force due to a change in velocity can be used to measure the velocity head in the Bernoulli equation. Care must be exercised to differentiate between these in setting up equipment, making readings, and analyzing data.

In general, pressures taken normal to the direction of fluid motion are static pressures. Static pressures can be observed in one of two ways: by making the observation through a small hole

in the container wall or by using a disc or tube type static head (Fig. 3.1). For observations requiring a high degree of açcuracy, particularly if there is a question as to the characteristics of flow, a number of small holes can be spaced evenly around the conduit in question and connected together by a manifold. This arrangement is called a piezometer ring. The disc and tube shown

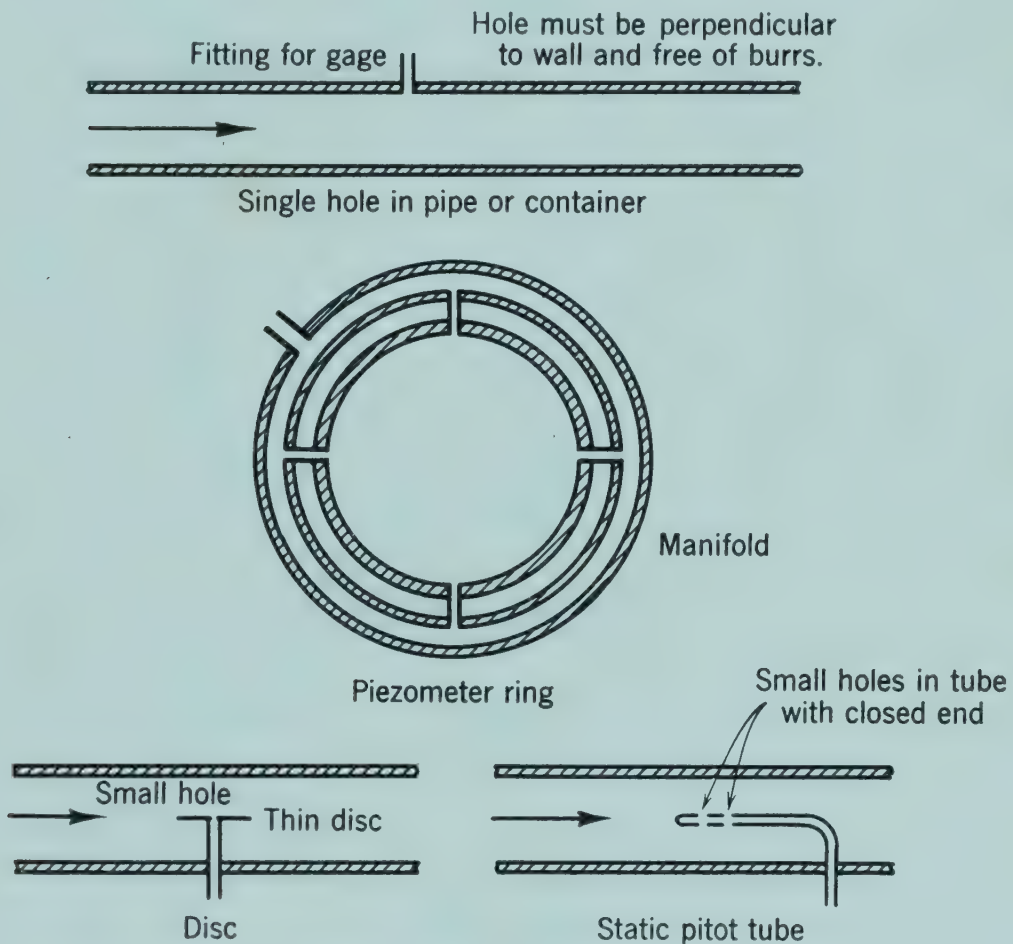


Fig. 3.1. Static pressure measuring devices.

in Fig. 3.1 can be used if installation of the other types are inadvisable or impractical.

The hole in the conduit must be perpendicular to the conduit wall and free of burrs. The conduit wall must be uniform in contour and smooth in the region of the hole. A hole $\frac{1}{8}$ in. in diameter may be used for small pipes $2\frac{1}{2}$ in. and under. For pipes up to 16 in., a $\frac{1}{4}$- to $\frac{1}{2}$-in. hole can be used. In any case, the smallest practical hole should be used; the smaller the hole, the greater the accuracy. Although single holes give reliable results if the velocity pattern in the conduit is symmetrical and the inside surface is uniform and smooth, the piezometer-ring type

of connection assures more reliable observations if flow properties or conduit characteristics are not ideal.

A simple but reliable piezometer for low-pressure measurement can be constructed as shown in Fig. 3.2. The holes should be spaced evenly around the tube. Four holes are recommended as

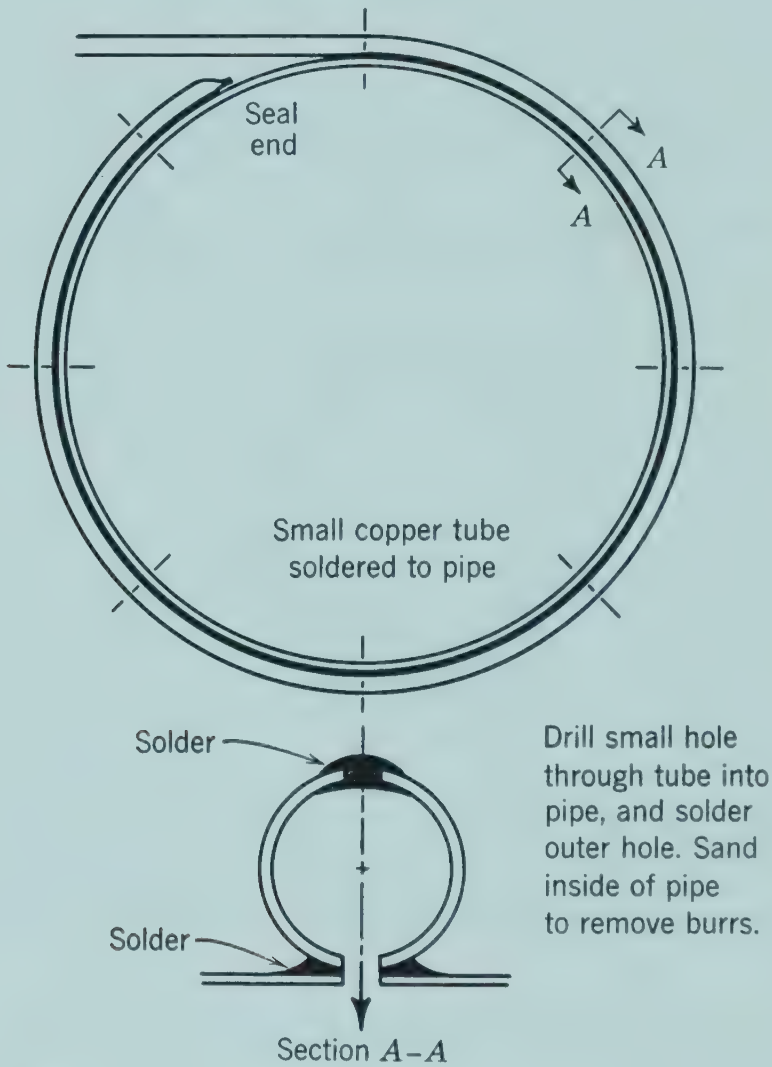


Fig. 3.2. A simple, effective piezometer ring.

minimum; six or eight would be more reliable. A large number of small holes are better than a small number of large holes.

PRESSURE GAGES

3.3. Manometers. The simplest and most reliable pressure gage is the manometer, which takes many forms in practice. Fig. 3.3 shows a few types which may be useful for the processing engineer.

The U tube is the simplest. The pressure is indicated by the difference in height of the tube columns in inches or feet of fluid contained in the manometer h if both sides of the U are filled with a gas.

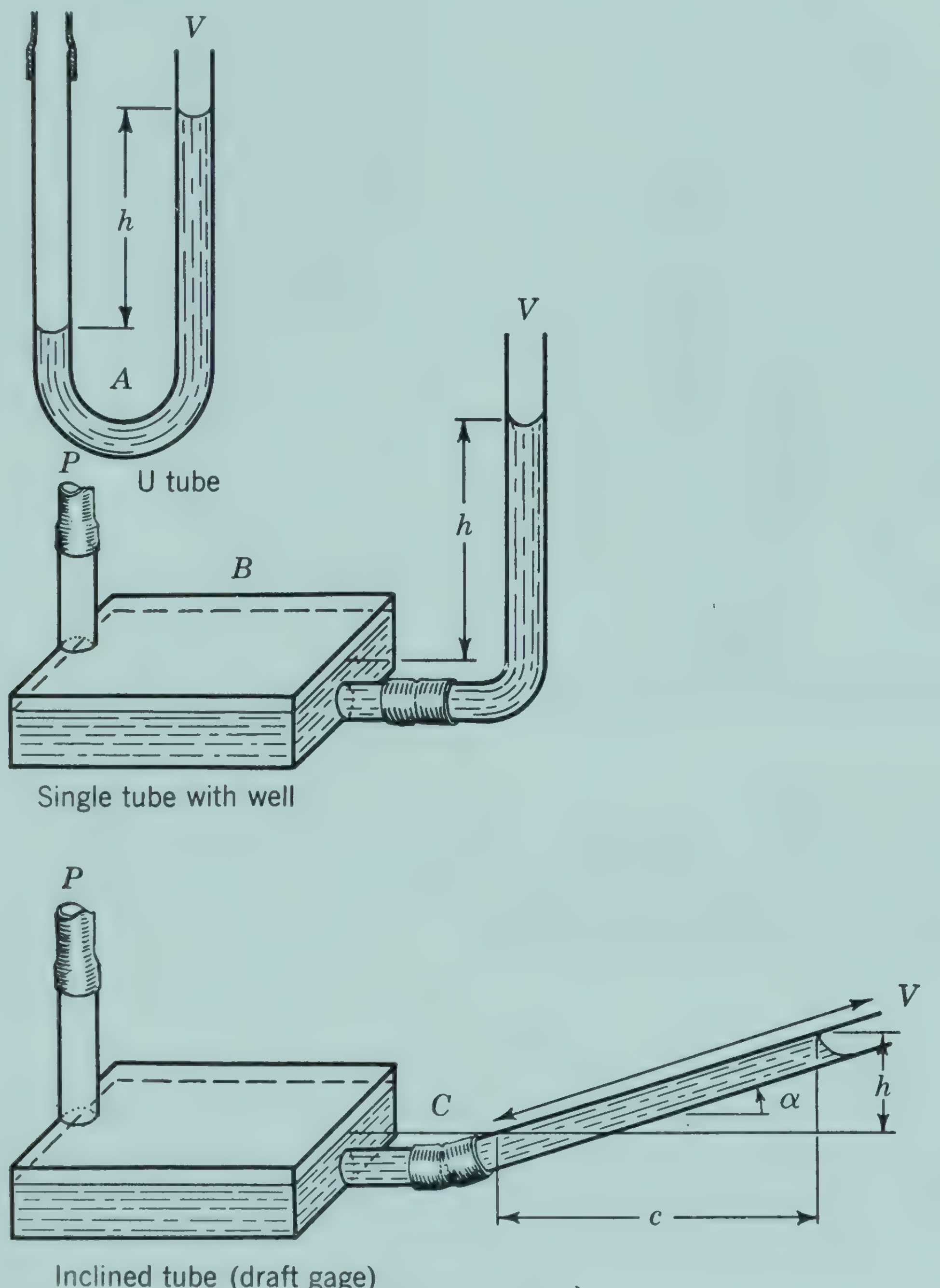


Fig. 3.3. Various types of manometers.

In general, any pressure gage used with a liquid must be located level with the desired point of pressure observation. If this is not done a significant error may result from a column of fluid rising in the connecting pipe.

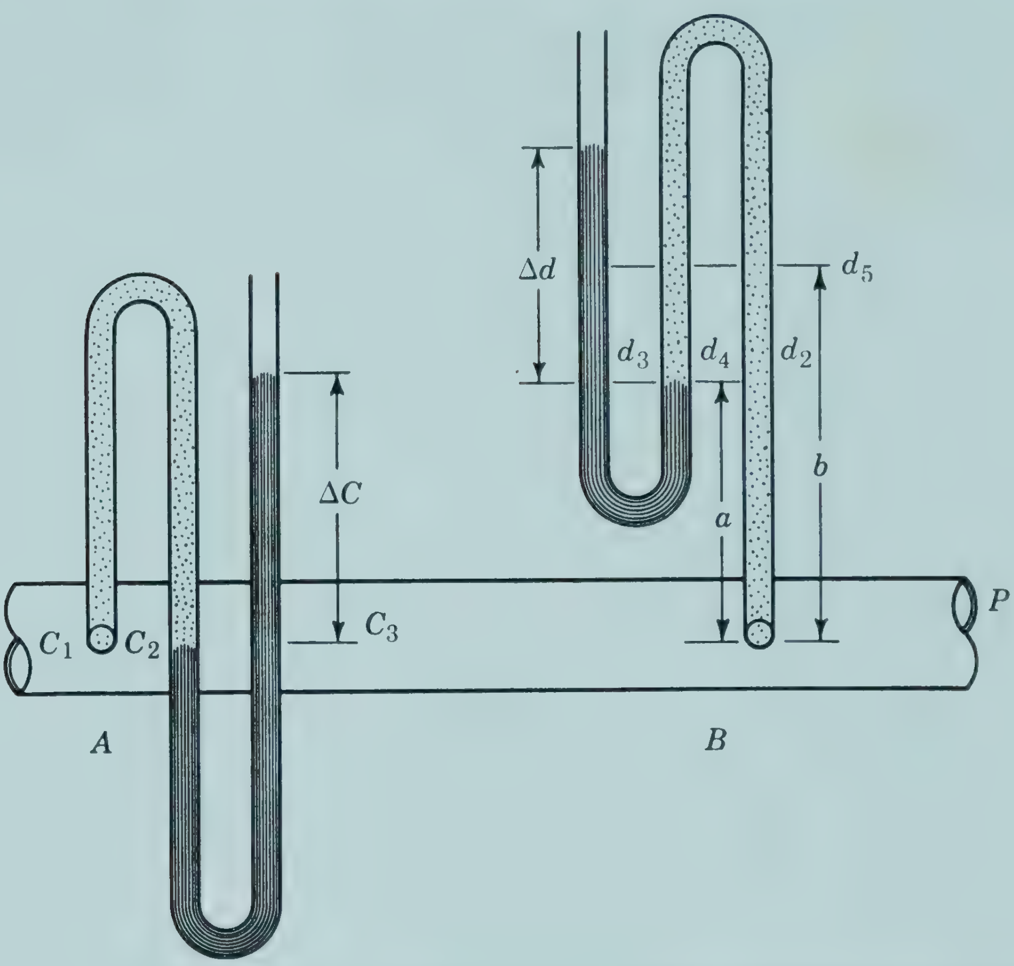


Fig. 3.4. U tube in line and above point at which pressure is to be measured.

Referring to A of Fig. 3.4, the differential Δc is a true measure of pressure since the force at $c_1 = c_2 = c_3$. The head for water at A is then, for a mercury tube,

$$H = 13.6\Delta c \tag{3.5}$$

the differential being measured in feet of mercury. However, as shown at B, the lower mercury meniscus is seldom level with the pipe axis. The force at $d_3 = d_4 = d_2$, and

$$H_{d_3} = 13.6\Delta d \tag{3.6}$$

The head at the pipe axis is greater than at d_2 by the height of the water column a. Since a changes with each change in pressure, it is often convenient to calculate the head at the average mercury level d_5, which is less than at d_3 by the height of the water column $0.5\Delta d$, d being measured in feet.

$$H_{d_5} = 13.6\Delta d - 0.5\Delta d = 13.1\Delta d \tag{3.7}$$

The head at the pipe axis is then

$$H = 13.1\Delta d + b \tag{3.8}$$

Where the mercury differential is measured in inches,

$$H = \frac{13.1}{12}\Delta D + b = 1.0917\Delta D + b \tag{3.9}$$

where ΔD = inches-of-mercury differential.

Equations 3.8 and 3.9 hold only when the connecting tube is completely filled with the liquid under consideration. If air is trapped in the tube, the liquid will rise only a short distance in the tube. The observed manometer pressure must be corrected by the elevation in the tube.

The single tube B (Fig. 3.3) has all the advantages of the U tube and none of its disadvantages. The well is large as compared to the tube so that the change in fluid level in the well is not significant for a fluid elevation in the column. Consequently, the manometer difference h can be read directly from a scale, no significant corrections being necessary. If desirable, the scale can be adjusted to compensate for variations in level of the well fluid thus giving a true reading.

The inclined tube C, also called a draft gage because of its use for observing chimney draft on furnaces, is a convenient means of increasing sensitivity. The scale multiplication or increase in sensitivity varies according to the following factor:

$$\csc \alpha \qquad \text{or} \qquad \frac{\sqrt{h^2 + c^2}}{h}$$

The inclination of the tube is limited by the surface tension characteristics of the fluid meniscus. When the inclination is too great, the meniscus has a tendency to "stick" and accurate readings are difficult. It is usually inadvisable to attempt multiplications of more than 20 with this type of gage. Because of irregularities in bore and straightness, these gages must be calibrated individually if a high degree of accuracy is desired.

The micromanometer in Fig. 3.5 can be constructed to read to 0.001 in. of fluid. It requires no calibration and can be constructed with a greater range than the inclined tube. It can be

used as a standard or for observations requiring a high degree of precision. The sensitivity is a function of the inclination of the glass tube carrying the cross hair. Versatility is made possible by permitting adjustment in the slope of the glass tube.

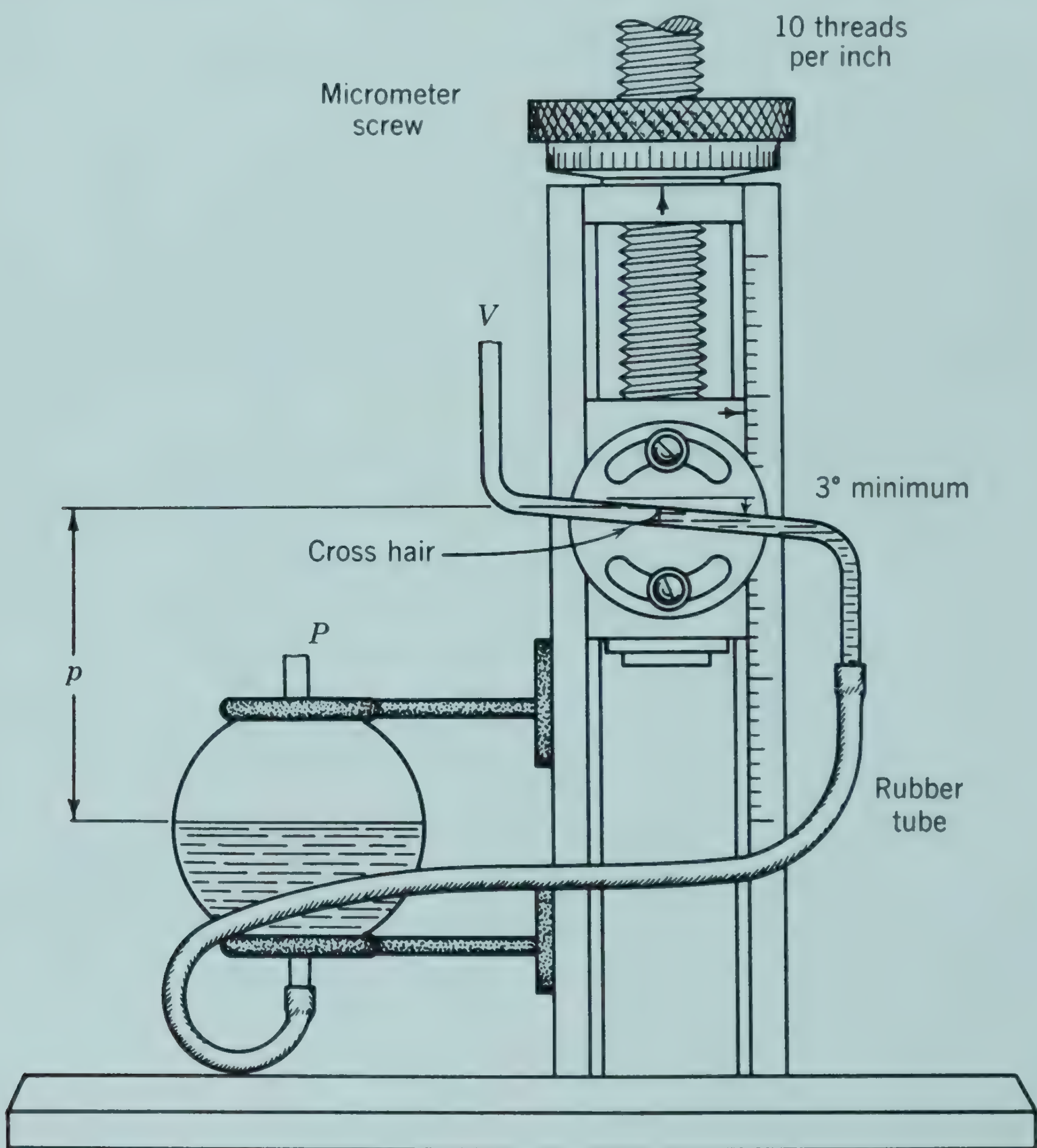


Fig. 3.5. A micromanometer that requires no calibration.

Alcohol has been found to be a good fluid for this and other manometers since its density and surface tension characteristics are superior to water.

All the gages shown in Figs. 3.3 and 3.5 can be used for observing pressures less than atmospheric by connecting to points *V*.

3.4. Bourdon Tube. The Bourdon tube type of gage (Fig. 3.6) is widely used for operation control where accuracies of approximately 2 per cent are acceptable, pressures are moderately high, and calibration does not have to be extremely consistent

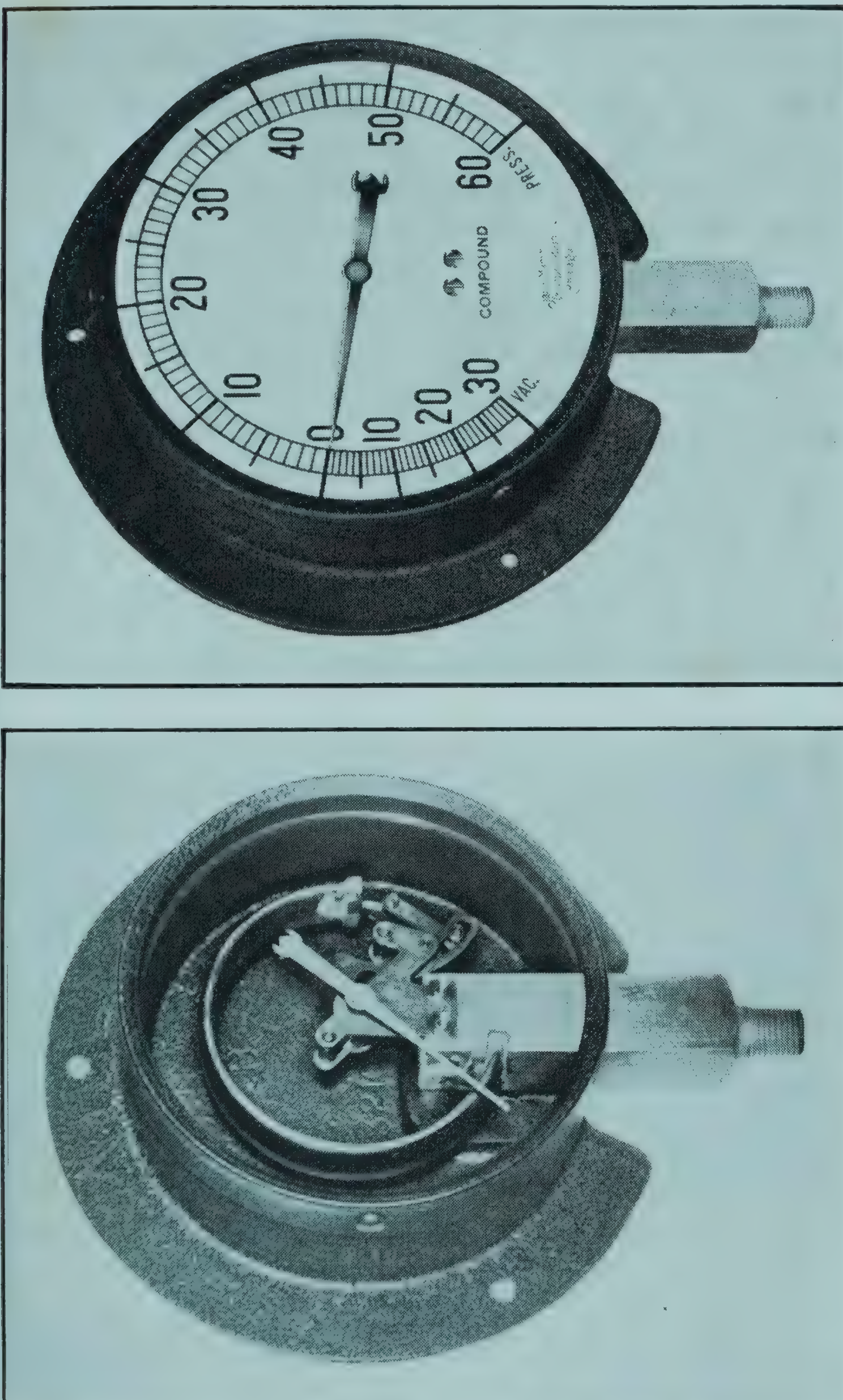


Fig. 3.6. Bourdon tube gage. Pressure in the circular shaped tube tends to straighten it out. The motion is transmitted through levers and gears to the hand. Pressure, less than atmospheric, vacuum, tends to contract the coil, and the indicating hand moves in the other direction. (*Courtesy* Jas. P. Marsh Corp.)

over the entire scale range. This is a secondary instrument since it must be calibrated against a known primary standard. Although gages of this type are, in general, not as accurate as certain other types of gages, most companies manufacture "test gages," which have a guaranteed accuracy of 0.5 per cent of full scale or better through the entire operating range. If care is exercised in their use, this accuracy can be maintained. This type of gage can be used for research, testing, and checking where a high degree of accuracy is mandatory.

In selecting Bourdon type gages, range of operation, temperature, type of fluid, accuracy, and operating condition must be considered. When steam pressure is being observed with a Bourdon type gage, a loop is used in the connecting pipe to form a water seal which prevents steam from entering the Bourdon tube. Designs are available for high-temperature operations, to withstand corrosive fluids, and to stand up under vibrating conditions. If properly selected, satisfactory performance can be assured.

3.5. Diaphragm. The diaphragm type of gage (Fig. 3.7) consists of a spring-loaded diaphragm or bellows which actuates

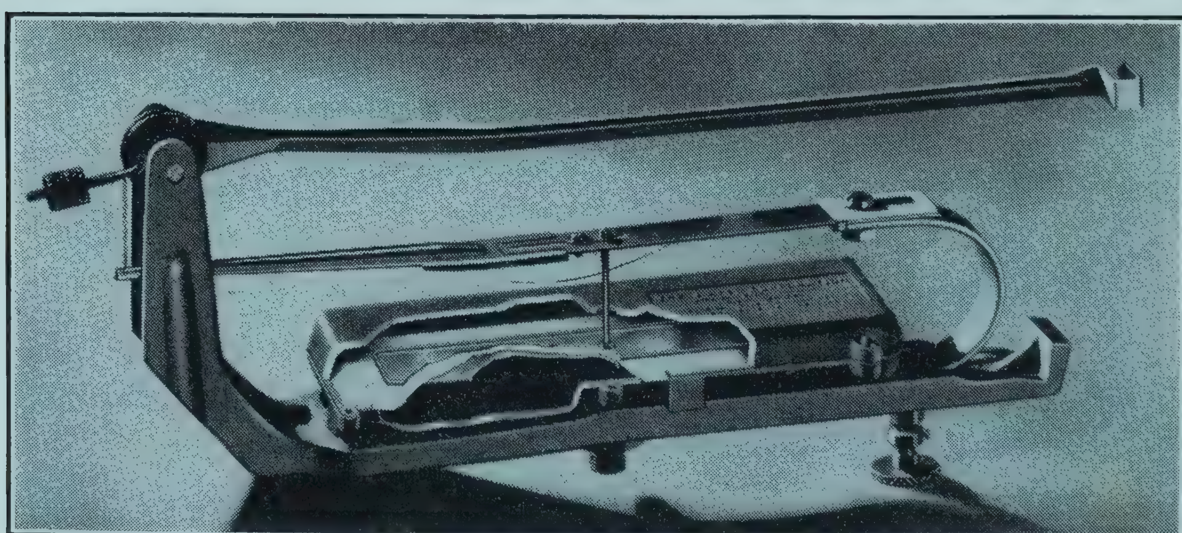


Fig. 3.7. Cutaway view of mechanism of a diaphragm type gage. (*Courtesy* The Hays Corp.)

a series of levers attached to the indicating hand. Gages of this type are designed for low-pressure operation, to 5 in. of water, approximately, and are nearly as accurate as the inclined water manometer or draft gage. These factors, plus mobility and ease

of operation, place this gage in a superior position to the inclined water manometer for many jobs.

3.6. Gage Throttling. Gages that are used for measurements where flow is fluctuating may vibrate or move otherwise, therefore accurate observations are impossible. Furthermore, the mechanical gages such as the Bourdon tube units may be damaged mechanically where excessive vibrations are encountered.

Gage installations on pumps, air and refrigeration compressors, and conduits where surging is experienced usually require protection against fluctuation. This is provided by placing a restriction in the gage line so the rate of flow through it is very slow. Adjustable valves, small orifices, or other restrictions that cut down the rate of flow but do not shut it off can be used. This procedure, which is called throttling, gives a true average of the varying pressures at the point where the gage is attached.

VELOCITY MEASUREMENTS

3.7. Pitot Tube. The pitot tube is essentially an open tube pointing into the stream of fluid flowing as shown in Fig. 3.8. The

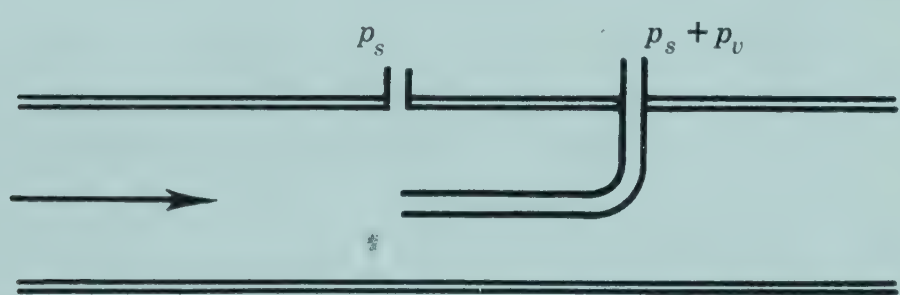


Fig. 3.8. Elementary pitot tube.

impact of the moving fluid creates a pressure head nearly equal to $V^2/2g$, which is the velocity head of the Bernoulli equation. The fluid static pressure or head which is made up of the pressure and elevation heads is added to the pressure head so that a pressure gage attached to the tube indicates the sum of the velocity, pressure, and elevation heads. A static pressure reading p_s is taken by the method described in a previous section; this reading is subtracted from the pitot total to give the net velocity or impact pressure. Pressure fittings are usually attached to a differential gage, and the impact reading is made directly.

Usually the static and total pressure elements are unified into a combined tube. Tubes of various design are available commercially. A recommended design is shown in Fig. 3.9. In general, the elementary type yields more reliable static pressure readings than the combined type because eddy currents may exist in the region of the static holes in the combined type. However, combined tubes designed and constructed on the basis of exhaustive tests will give results well within accepted engineering tolerances without the use of a correction factor.

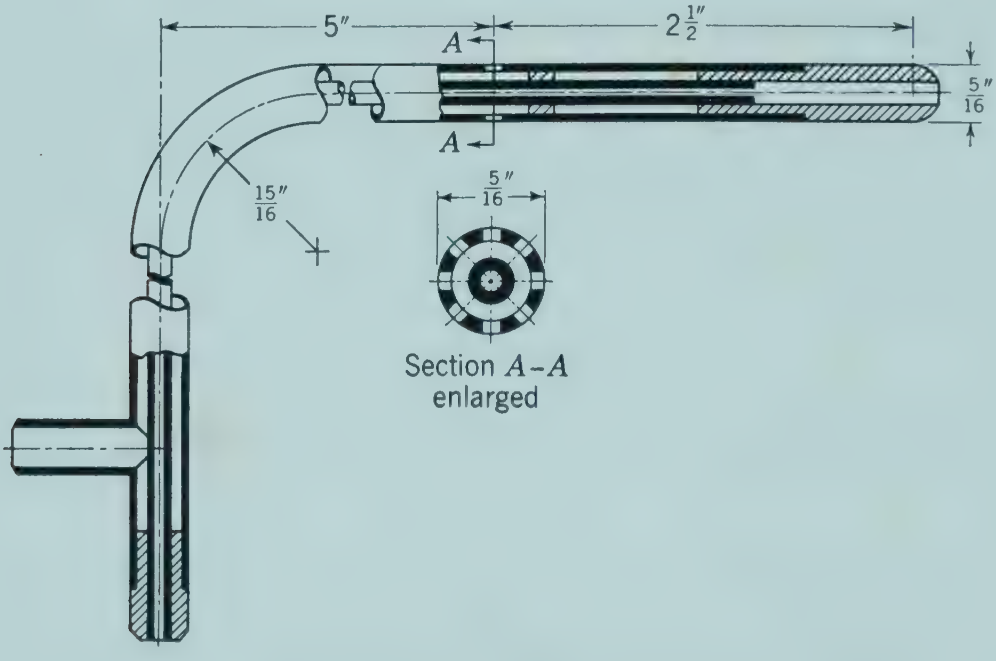


Fig. 3.9. A practical and efficient pitot-static tube.

The head based on the Bernoulli theorem and indicated by the net or differential pitot pressure is nearly

$$V^2/2g = H \tag{3.10}$$

H being in feet of the fluid flowing. The presence of the stem in the air stream causes an increase in the static reading and a decrease in the velocity pressure. Since the observational errors are usually greater than the known performance error, usual practice is to disregard the errors and assume that no correction factor is needed. This can be done without significant error resulting. It is conventional to express pitot pressures in pounds per square inch for liquid flow and in inches of water for gases. These conversions follow.

$$H\gamma/144 = p \tag{3.11}$$

substituting for H in equation 3.11 and solving for V gives

$$V = 96.4\sqrt{p/\gamma} \tag{3.12}$$

which holds for any fluid of specific weight γ and gives the velocity in feet per second for net pressures in pounds per square inch. The velocity in feet per second for pressures in inches of water is

$$V = 18.3\sqrt{p/\gamma} \tag{3.13}$$

These equations (3.12, 3.13) give the true velocity of the fluid at the tip of the tube only and do not indicate the average since

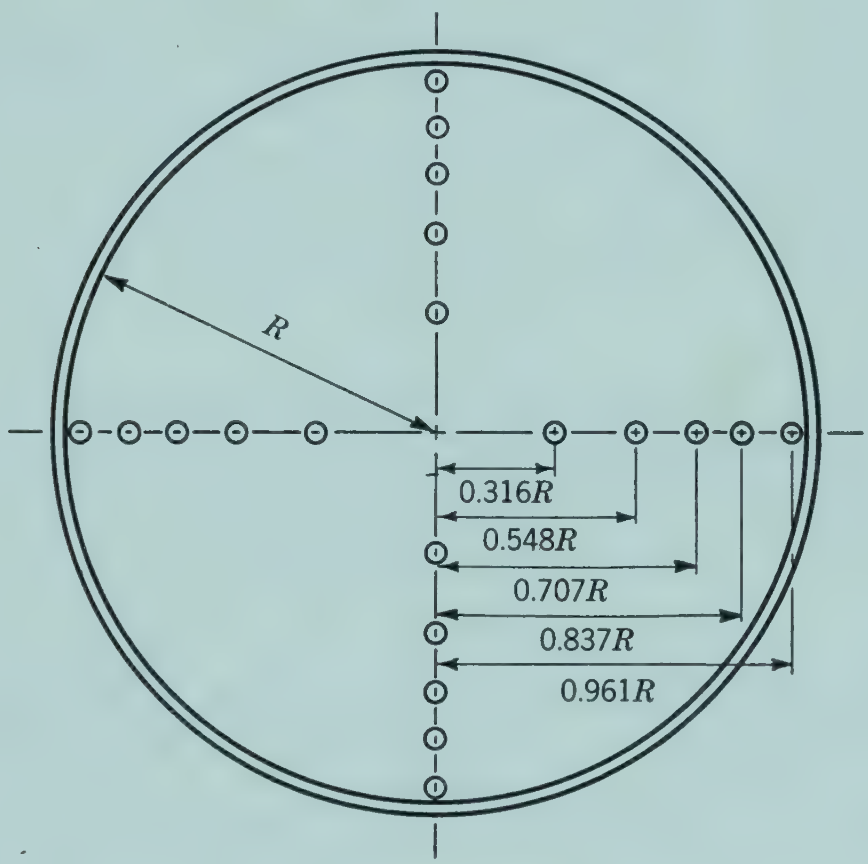


Fig. 3.10. Pitot traverse points in a round duct.

it is known that the velocity is a maximum at the center of a conduit and decreases toward the walls. Average velocities can be determined by dividing the conduit into a number of small equal concentric areas, observing the velocity at the center of each area, and finding the average of these. A system for doing this is shown in Fig. 3.10. If the installation is a permanent one, the average velocity can be determined for a number of velocities through the normal operating range and these velocities can be referred to the maximum velocity at the center by a factor so

that the single center reading will indicate the true average. A factor 0.85 to 0.97 times the maximum usually indicates the true average for turbulent flow. Observations in a long pipe will be more consistent through a high range of velocities, although the factor will be low. The factor is 0.5 for streamlined flow.

Observations should be made in the middle of a long straight uniform section of pipe. For best results, the length should be at least 20 times the diameter. Observations close to bends, outlets, blowers, grills, etc., will probably give erroneous results.

Although the pitot tube performs properly at very low velocities, the sensitivity of the manometer gage with which it is usually used is insufficient to measure velocities of under 400 ft per min with the requisite accuracy. The micromanometer (Fig. 3.5) can be used for somewhat lower velocities, but other methods of measurement will provide more acceptable results.

3.8. Venturi Meter. The venturi meter shown in Fig. 3.11*a* is preferable to the pitot tube when average cross-sectional veloci-

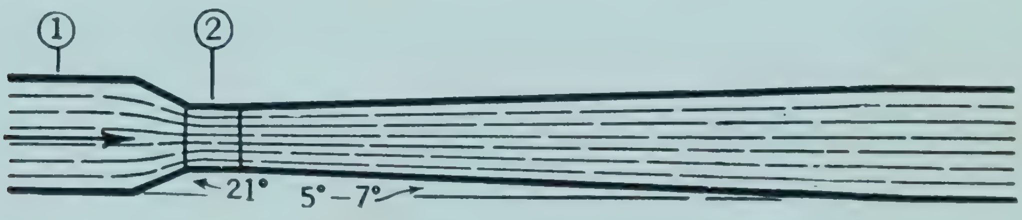


Fig. 3.11*a*. Venturi meter.

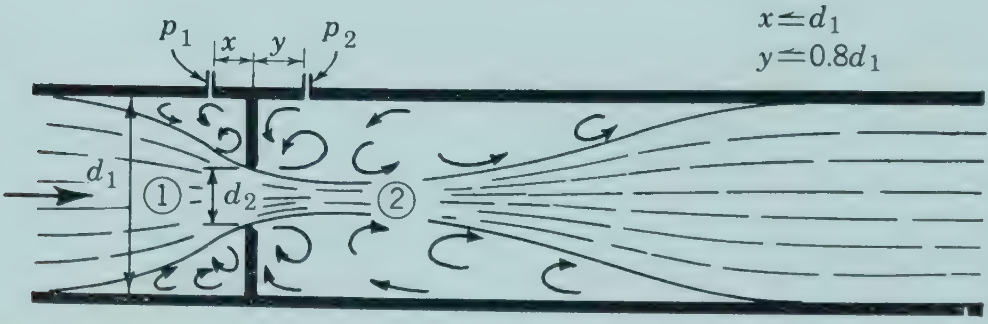


Fig. 3.11*b*. Orifice meter.

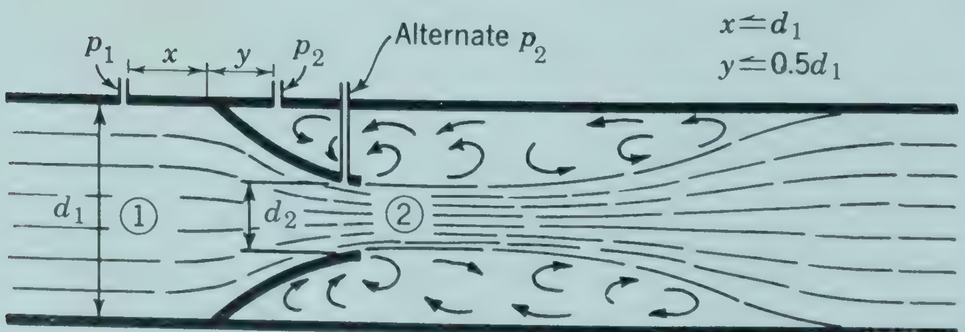


Fig. 3.11*c*. Nozzle.

ties are desired. The velocity indicated is a true average, and the pressure difference can be magnified by increasing the diameter ratios so that more accurate readings can be obtained. It is an excellent measuring device for permanent installations, but, because of its bulk and the fact that it is an integral section of the conduit system, it is not readily mobile.

Considering the Bernoulli equation from point 1 to 2, Fig. 3.11*a*, note that only the velocity and pressure heads are effective and

$$\frac{144p_1}{\gamma_1} + \frac{V_1^2}{2g} = \frac{144p_2}{\gamma_2} + \frac{V_2^2}{2g} \tag{3.14}$$

The equation of continuity is

$$A_1V_1 = A_2V_2 \tag{3.15}$$

and

$$V_2 = (A_1V_1)/A_2 = nV_1 \tag{3.16}$$

where

$$n = A_1/A_2 = d_1^2/d_1^2 \tag{3.17}$$

Substituting this value of V_2 in equation 3.14, V_1 is found to be

$$V_1 = C\sqrt{2g144/(n^2 - 1)(p_1/\gamma_1 - p_2/\gamma_2)} \tag{3.18}$$

C is a coefficient that corrects for energy loss resulting from eddies and friction in the meter and is usually 0.95 or more. Values of C have been determined experimentally and are shown graphically in Fig. 3.12. In properly designed venturii the error resulting from the use of this graph is not expected to be greater than 2 per cent.

Note from equation 3.18 that an increase in V_1 or an increase in n will cause a decrease in p_2, assuming that p_1 remains constant. With liquids, if p_2 drops as low as the vapor pressure of the fluid, vaporization will occur at any slight irregularity. The formation and subsequent collapse of vapor bubbles promotes erosion of metal. This process, called cavitation, not only limits the venturi as a measuring device but also causes an increase in energy loss and erosion or pitting of the tube itself.

Although cavitation must be avoided when a venturi is used as a measuring device, the phenomenon which produces it is used

in certain types of pumps. If V_1 and n are sufficient to produce a p_2 less than atmospheric, this rarefied pressure, vacuum, can be used for evacuating or pumping. The laboratory suction pump that is fastened to a water faucet and basement sump pumps operated off the house water systems are good examples of venturi pumps.

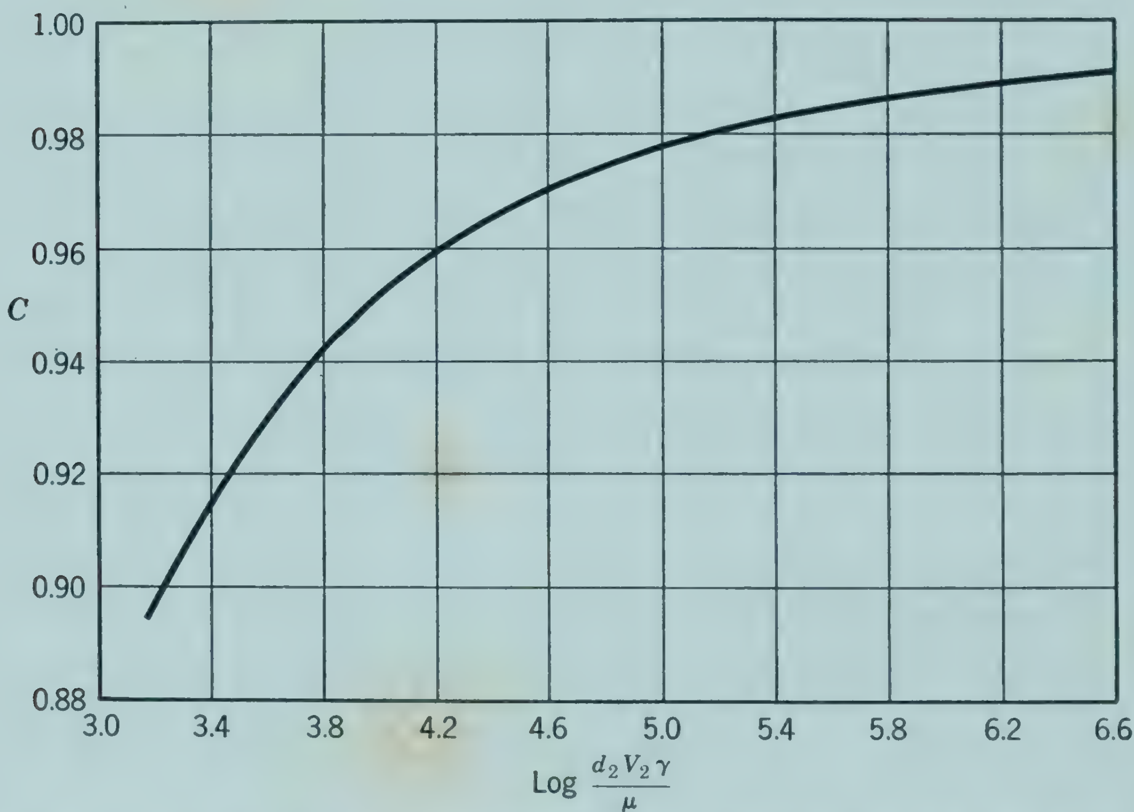


Fig. 3.12. Values of the venturi coefficient C referred to the log of Reynolds number.

The previous discussion applies to liquids where $\gamma_1 = \gamma_2$. If γ_2/γ_1 is nearly equal to 1, gases may be considered incompressible and the error resulting is negligible. Ratios of 0.95 and 0.90 produce errors of 4 and 6 per cent respectively. Since it is advisable to calibrate an individual tube against a known standard for most accurate results, preliminary calculations for gases can be made with equation 3.18 without serious error resulting. Ventilation, drying, and air conditioning pressures with which the processing engineer is active seldom will exceed 4 in. of water. The ratio for this pressure referred to atmospheric at 14.7 psi is 0.99, the resulting error being 0.5 per cent.

3.9. Orifices. Orifices and nozzles, Fig. 3.11*b* and 3.11*c*, are convenient devices for measuring rates of flow because they are

simply constructed, easily installed, and occupy little space as compared to the venturi. Although commercial units are available, shop-made meters give reliable results. Orifices and nozzles are subject to considerable energy loss due to eddies and friction and are inferior to venturi in this respect. Nozzles and orifices have approximately the same head or energy loss resulting from turbulence, but, for a given flow, the differential head for a nozzle is less than for an orifice. These meters are convenient for measuring or calculating discharges into the air and into or out of large bodies such as storage tanks since a hole or valve is essentially an orifice.

Equation 3.18 derived for the venturi meter also applies to orifices and nozzles. Considerable care must be used in selecting values for the coefficient C, which varies widely with n, the type of orifice, and to some extent with the location of the pressure taps.

If the pipe to the right of the orifice in Fig. 3.11*b* were removed, the diameter of the stream at point 2 would be smaller than the orifice. This contraction, called the vena-contracta, results because the fluid is unable to make an abrupt turn past the edge of the orifice. In a closed pipe, considerable turbulence results and there is a marked energy loss. This loss plus the variation between the size of the orifice and the minimum diameter of the vena-contracta, produces low values of C in equation 3.18 for certain conditions that will be discussed later. The nozzle is designed to gradually bend the fluid so that the discharge stream does not contract materially. More satisfactory discharge coefficients result.

The location of the pressure taps is shown in Fig. 3.11*b*. For most reliable results, a piezometer is recommended. The size and characteristics of the holes should follow the recommendations for static-pressure observations given in sect. 3.2. The actual location of the pressure taps may alter the flow coefficient C, but this is immaterial since each unit should be individually calibrated unless it has been carefully constructed on the basis of recommendations by the A.S.M.E., *Power Test Codes*.

Discharge coefficients vary with the density and viscosity of the fluid, the speed characteristics of flow, instrument dimension, and pipe roughness. The coefficient C for nozzles is approximately 0.97 ± 0.02 and in general varies from 0.60 to 0.80 for

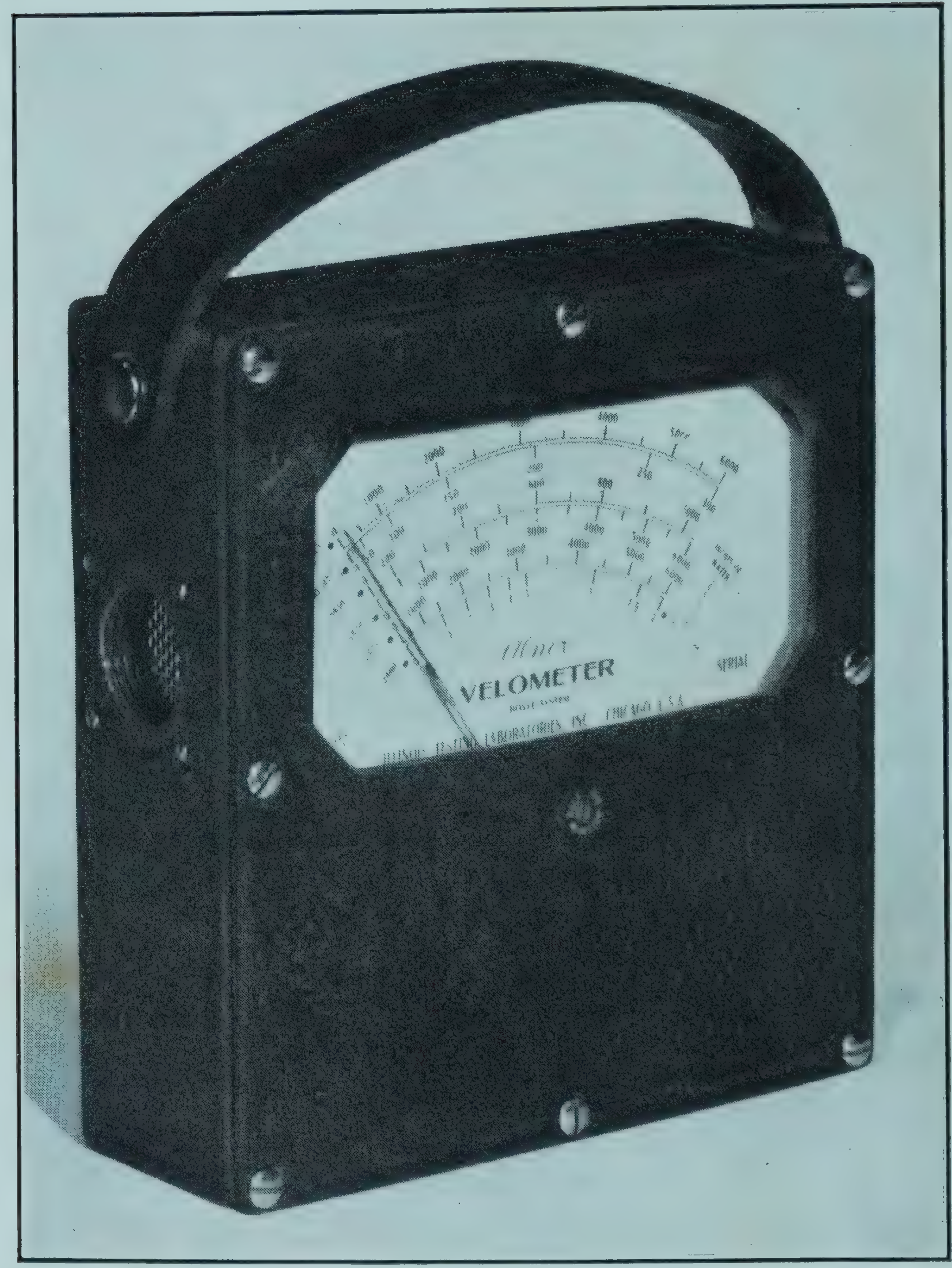


Fig. 3.13. A commercial swinging vane meter for measuring air flow. The instrument faces into the stream of air. The air moving into the grilled port activates a spring-loaded vane which is attached to the indicating hand. The cross-section drawing shows the operating principle and the method used for making remote readings. (*Courtesy* Illinois Testing Laboratories, Inc.)

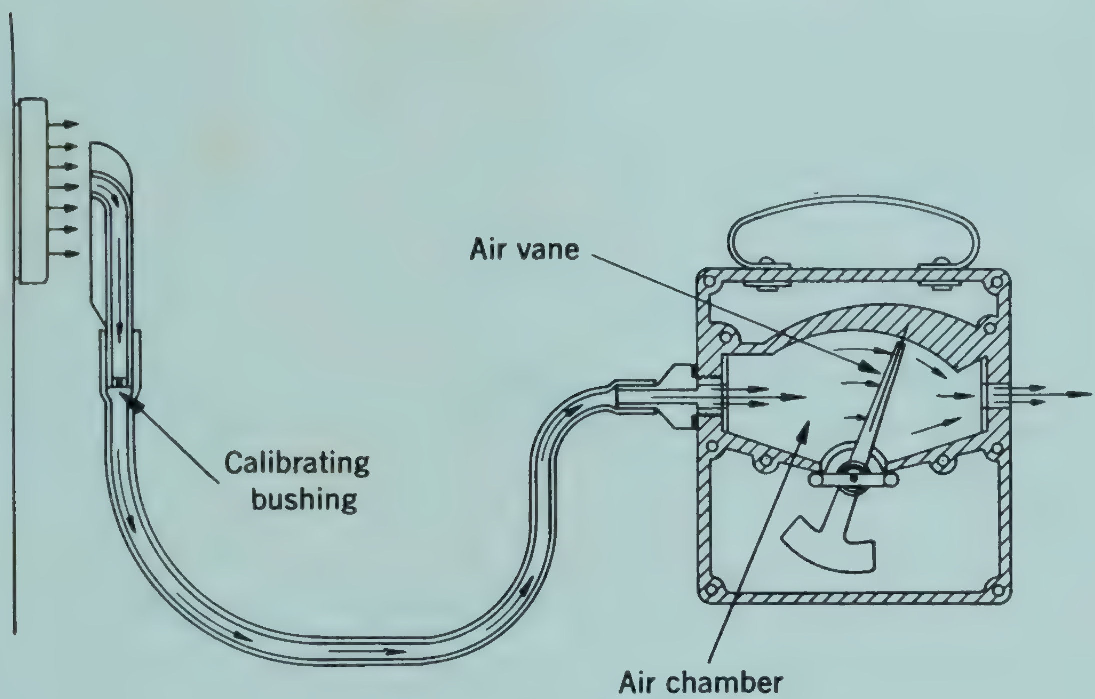


Fig. 3.13 (*continued*)

orifices with sharp edges. For a complete set of coefficients consult the A.S.M.E. Power Test Codes.[2,7]

3.10. Rotating-Vane Anemometer. Instruments of this type are essentially small windmills which indicate the linear air travel through them. The average velocity for a time t is determined by dividing the distance by the time x as observed by a stop watch. Each instrument must be calibrated individually. They operate satisfactorily for velocities of 5 to 50 ft per sec. Special instruments are available for lower velocities, but extreme care must be used in handling and maintaining them to insure continued accuracy.

When making a test, the instrument must be reasonably well aligned in the direction of air motion. Two or more individual determinations should be made to determine the variation in rate of flow and to insure a more reliable average value. No individual reading should be made for a time of less than 1 min or a linear reading of 100 ft.

3.11. Swinging-Vane Meter. The swinging vane meter is essentially a spring- or gravity-loaded gate which is moved by the impact of the flowing fluid. The formula showing the performance of this type of meter is complicated, and the characteristics of flow past the vane are difficult to rationalize. Consequently, meters of this type are usually based upon laboratory tests. A commercial swinging-vane meter is shown in Fig. 3.13. It is

fitted and calibrated so that high and low air velocities and static pressures can be observed both directly and remotely. Static-pressure observations are possible since a certain static pressure operating through a definite resistance will produce a definite rate of flow through the instrument. The pressure-rate-of-flow relationship is determined, and the instrument is also calibrated in terms of pressure.

The degree of accuracy of the instrument depends upon the precision of manufacture, calibration accuracy, and care in operation and handling.

LOW-VELOCITY MEASUREMENTS

Low velocities, which for convenience are designated as less than 300 ft per min (5 ft per sec), may be difficult to measure, particularly those of gases.

Fortunately, the measurement of liquid velocities at low values can usually be made easily and accurately. The devices discussed under high velocity measurements operate satisfactorily somewhat within the above-designated range because the specific weight is high enough to provide sufficient kinetic energy to operate the gages used. Below this point, flow meters and gravimetric procedures may be used with suitable results because of the relative incompressibility of the fluid, high density (as compared with gases), and ease with which fluids can be confined.

Measurements of gas velocities below 300 ft per min are difficult because the kinetic energy involved is insufficient to activate the gages used. Volumetric meters perform satisfactorily under certain conditions, but their use is limited. Natural convection currents produce an additive error at low velocities which may completely confuse the results. In spite of the recognized difficulties, fairly reliable techniques have been developed for measuring gas velocities, particularly air. A discussion of these techniques follows.

3.12. Katathermometer. The katathermometer is a large-bulb alcohol thermometer of specific dimensions with 95° and 100°F gradations only. The thermometer is heated in a water bath or by other suitable means to a point above the 100° mark. After the bulb is carefully dried, the thermometer is placed in the air stream to be measured, and the time required for the bulb

temperature to drop from 100° to 95° is noted by a stop watch. The rate of cooling is a function of surface conductance, which in turn is a function of the air velocity past the bulb. The mathematics of the katathermometer as given by Severns [9] follow:

$$V = 3.28\left[\frac{(F/\theta) - 0.13t_c}{0.47t_c}\right]^2 \tag{3.19}$$

For velocities above 3.28 ft per sec

$$V = 3.28\left[\frac{(F/\theta) - 0.2t_c}{0.4t_c}\right]^2 \tag{3.20}$$

For velocities below 3.28 ft per sec

t_c = 36.5° minus room temperature, °C.
F = thermometer factor, by calibration.
θ = time in seconds for drop from 100° to 95° F.

Note that the time of response is an inverse function of the temperature difference t_c for any one velocity. Consequently, the over-all accuracy is dependent upon the ability of the operator to make accurate observations of both temperature and time.

Velocities of air at temperatures above 100°F can be measured by cooling the bulb below 95°F and noting the time required for the temperature to rise the required 5°F, the time being given a negative sign.

3.13. Hot-Wire Anemometer. The hot-wire anemometer, Fig. 3.14*a*, is based on the variation in resistance of an electrical conduit with conduit temperature and the variation of the conduit temperature with the velocity of a gas past the wire.

A small platinum wire 0.004 in. or less in diameter and 2 in. long is heated by an electric current to a high temperature. The high-temperature wire is cooled by the motion of air past it. Since the resistance of the wire varies with temperature, the amount of current flowing will vary with the velocity of air past the heated wire. An increase in velocity will permit an increase in the current flowing since the cooled wire will offer less resistance to electrical flow.

The current required to maintain the wire at a prescribed constant temperature and constant resistance when the air has a velocity V is

$$i^2 = i_0{}^2 + k\sqrt{V} \tag{3.21}$$

where i_0 is the required current at zero velocity. Two methods of measurement are used: the constant-resistance method and the constant-current method.* In each method, the hot wire is placed as one arm of a Wheatstone bridge with suitable gages and controls to adjust and observe operating conditions. The hot-wire anemometer in Fig. 3.14*a* is wired to operate at con-

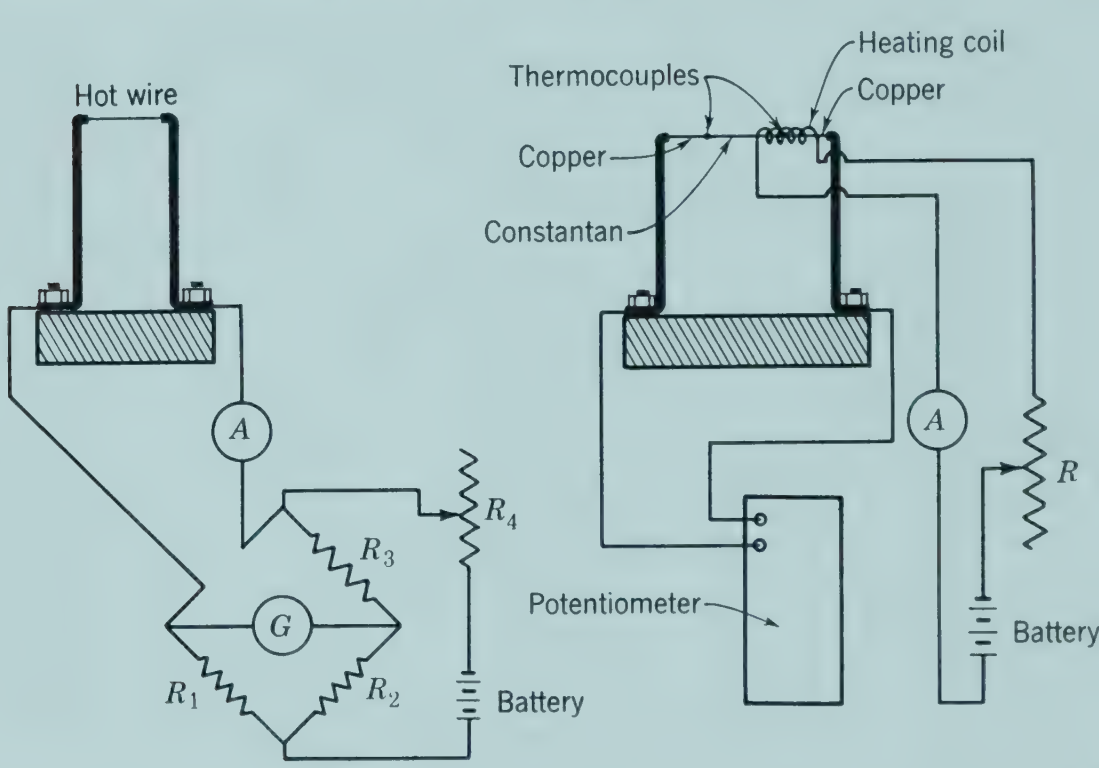


Fig. 3.14. Hot-wire anemometer, left, and thermocouple anemometer, right.

stant resistance. The resistances R_1, R_2, and R_3 of the Wheatstone bridge are adjusted so that the resistance of the hot wire can be held at a constant prescribed value by adjusting the variable resistance R_4. The velocity, then, is related to the current according to equation 3.21. The null reading i_0 is taken by covering the hot wire with a small container which is assumed to produce still air. Convection currents set up by the heated wire produce local air movement which makes the results questionable. If the constant resistance-method is used, k can be determined for a single air velocity and the calibration curve can be calculated. The instrument is calibrated by attaching it to the end of a long arm which is rotated in a room of still air. The peripheral speed of the end of the arm at which point the meter

* Refer to Ower [6] for a detailed study of these methods.

is attached is considered the rate of air movement. This procedure must be conducted with care else convection currents produced by cold windows, walls, lights, radiators, motors, the operator, etc., will cause poor results, especially at low velocities.

3.14. Characteristics of Hot-Wire Anemometer. The hot-wire anemometer can be used for velocities as low as 6 ft per min by using a wire temperature in the order of 160°F. Wire temperatures are increased for measuring higher velocities, 1800°F being used for velocities in the order of 800 ft per sec.

Free convection around the heated wire introduces a constant directional error which may amount to 30 ft per min for a 0.003-in. wire at 1800°F, and 15 ft per min at 400°F. Consequently, if low velocities are to be measured, the instrument must be calibrated under the same conditions as those under which the tests will be made or suitable correction procedures must be followed.

Heat energy is lost from the hot wire by radiation, but no correction need be made in this respect under normal conditions because of the small area involved.

A significant error will result if the ambient air temperature is significantly different from the ambient air temperature under which calibration was carried out. The lower the wire temperature, the greater will be the error for a constant change in ambient air temperature. Corrections can be made for this variation, but the procedure is too involved to be discussed here.

Although the hot-wire anemometer can be used at low-wire temperatures for measuring very low velocities, it performs best under conditions of moderate to hot temperatures for measuring moderate to high velocities.

The direction of air motion must be known so that the hot wire can be located perpendicular to it. If the wire is placed at an angle to flow, low-velocity indications result. Corrections cannot be made under this condition since the relationship between the adjustment factor and angle is not known.

Fluctuating velocity is difficult to observe since the equipment must be balanced for each velocity. Automatic adjusting and recording equipment can be used under fluctuating temperatures if the high cost can be justified.

This measuring device can be used in a small space and operated and observed from a remote location.

3.15. Thermocouple Anemometer. The thermocouple anemometer shown in Fig. 3.14*b* operates on the same basic principle as the hot-wire anemometer. A predetermined standard current is passed through the heating coil. This raises the temperature of the enclosed thermocouple. Air moving past the heated coil cools it. The cooling effect is reflected in the difference in temperature between the thermocouples. Consequently, the air velocity is related to the potential between the thermocouples. The potential across the thermocouples is nearly a linear function of the temperature difference and can be observed by a potentiometer. However, since it operates on the Wheatstone-bridge principle, no current is flowing when a reading is made. Consequently, the size and length of leads, if reasonable, do not affect performance.

The basic equation that relates the factors involved in the thermocouple anemometer is

$$\sqrt{V} = \frac{i^2C}{t_1 - t_2} \tag{3.22}$$

in which V is the air velocity; i is the heater element current; and t_1 and t_2 are the temperatures of the hot and cold thermojunctions respectively. C, an empirical constant, is composed of the resistance value of the heating element, heat lost by radiation, the coefficient of thermal conductance of the heating element and the wires to which it is attached, and a proportionality constant. Heat loss by radiation is recognized as existing, but its effect appears to be insignificant. Natural convection past the heated element produces an indicated velocity when the ambient air velocity is zero. This effect appears to be nullified when velocities are in the order of 5 ft per min or higher.

The thermocouple anemometer can be calibrated by the same method as previously described for the hot-wire anemometer. The difficulties to be overcome and shortcomings are the same as experienced when calibrating a hot-wire anemometer.

Brooks of the California Agricultural Experiment Station substantiated the basic equation by laboratory tests.* Consequently, a performance curve can be secured by calculating the value of C from the operating data for a single velocity.

3.16. Characteristics of Thermocouple Anemometer. The thermocouple anemometer is superior to the hot-wire anemometer,

* Unpublished.

especially for very low-velocity readings. The hot junction is maintained initially at 15°–20°F above air temperature, whereas the hot-wire anemometer must operate at 60 to 1700 degrees above air temperature. Radiation and convection losses are small relative to the thermocouple anemometer as compared to the hot-wire anemometer, which operates at much higher temperatures. A normal change in air temperature does not affect the performance of the thermocouple anemometer since the heating element is made of Manganin, which has a constant resistance through a wide temperature range, and the two thermocouples are referred to ambient air temperature.

The thermocouple anemometer is affected by direction of approach of the air. Fluctuations in velocity may create a reading difficulty if an automatic recorder is not used.

The thermocouple anemometer, like the hot-wire anemometer, can be placed in a small space and operated from a remote location.

3.17. Thomas Meter. This measuring system is based upon the rise in temperature that results from the introduction of heat into a confined stream of flowing fluid that can be either liquid or gaseous. An electrical heating element is placed in the stream of flowing fluid and raises the temperature of the fluid.

The relationship between the heat energy added, the temperature elevation, and rate of flow is shown by the following formula.

$$V = \frac{0.0569 \text{ volt} \times \text{amp}}{A\gamma c_p(t_1 - t_2)} \tag{3.23}$$

where V = fluid velocity, ft per min.
A = cross-sectional area, sq ft.
γ = specific weight of fluid, lb per cu ft.
c_p = specific heat of fluid, Btu per lb °F at constant pressure.
t_1 = upstream or cool temperature, °F.
t_2 = downstream or hot temperature, °F.

Temperatures are observed by resistance thermometers or thermocouples. They are connected to a milliammeter or potentiometer which indicates small temperature differentials accurately. The heating-element current is adjusted to maintain a constant temperature differential, usually of 2° to 5°F. Note that the velocity is proportional to heating-element amperage

when a constant temperature differential is maintained. Small temperature differentials are advisable in order to minimize the effect of heat loss by radiation and conduction through the walls. Losses by radiation can be eliminated by shielding the heating element. Conduction losses can be eliminated by insulating the conduit containing the metering elements. Automatic recorders can be set up by activating the heating-element rheostat through the action of the galvanometer.

3.18. Method of Mixtures. Rates of flow in confined conduits can be determined by metering a foreign fluid into the stream at a definite rate and sampling the mixture at a point downstream where complete mixing is assured. If an inert gas is metered into a stream of flowing air at atmospheric pressure, the percentage of gas in the mixture as sampled indicates the rate of flow.

The percentage of gas in the mixture is

$$Pg = \frac{Wg}{VA\gamma + Wg} \tag{3.24}$$

from which

$$V = \frac{Wg}{A\gamma}\left(\frac{1}{Pg} - 1\right) \tag{3.25}$$

where V = rate of air flow, ft per min.
Wg = gas added, lb per min.
A = area of conduit, sq ft.
γ = specific weight of air, lb per sq ft.
Pg = percentage of gas in sampled mixture expressed as a decimal.

In practice, the amount of gas added should be adjusted so that the sample of gas mixture can be analyzed accurately. A comparable procedure can be used for liquids by introducing a standardized salt solution into the flowing stream at a constant rate. The concentration of the salt in the sample is determined by titration or other suitable quantitative method.

3.19. Ammonium Chloride. The ammonium chloride generator shown in Fig. 3.15 can be used to observe and measure the movement of air in a confined space such as a room, cold storage locker, or storage house.

The motion of the dense cloud of ammonium chloride which is ejected into the atmosphere by operating the bulb will travel

with the air into which it is discharged. The linear speed and direction can be observed.

The rate of settling of the particles is so slow that they may be considered as moving perfectly with the air. This method is superior to that using smoke since there is no heat evolved to

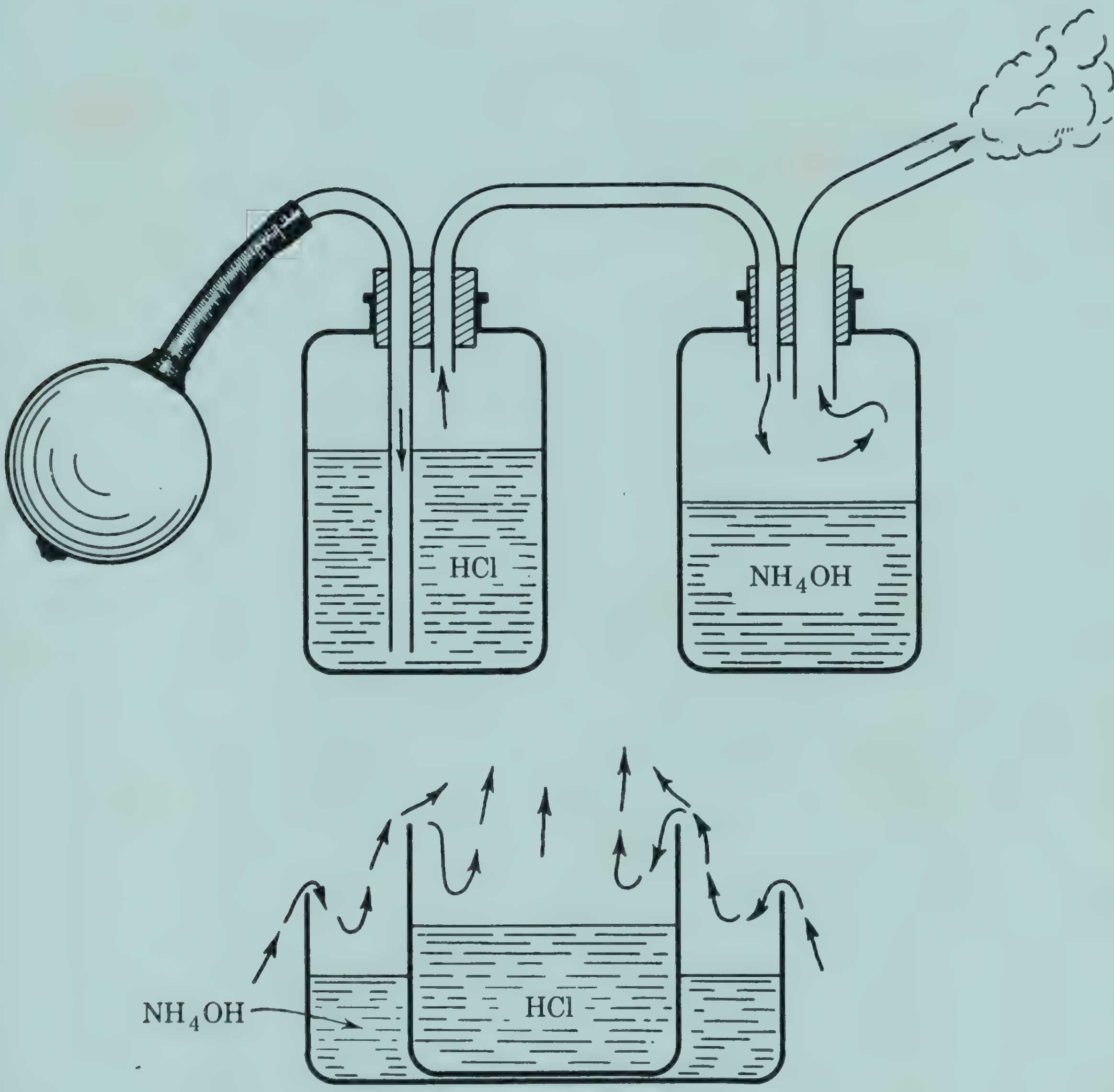


Fig. 3.15. Ammonium chloride generators.

produce convection. Best performance can be expected if the device is operated by remote control, by extending the aspirator-bulb tube. Thus, the body heat of the operator does not affect the rate of air movement.

FLOW MEASUREMENT

Flow implies a quantity, cubic feet, gallons, etc., flowing per unit of time as compared to velocity which implies linear rates.

The basic flow-measurement procedures are by weight and volume per unit of time. Fluids flowing at a constant rate may be run into or out of a container that is weighed at the beginning and end of a time period. The net weight increase divided by the time provides the rate. There are comparable procedures which use volume.

Any of the velocity indicating or measuring devices discussed in the previous section can be used to indicate flow by multiplying the cross-sectional area by the average velocity. For permanent installations where the effective area is constant, gages and other indicating devices are frequently calibrated to indicate rates of flow directly. Recording indicators are sometimes used to indicate the quantity accumulated or amount passed during a specified time.

Many other types of flow meters are available commercially. They are complete units designed for specific installations and conditions. The more common types are discussed in the following sections.

The following factors should be considered when selecting a meter for a specific job.

A. Operating conditions.
   1. Characteristics of material to be metered.
   2. Operating range.
   3. Line pressure.
   4. Characteristics of flow, steady or surging.
   5. Required accuracy.
B. Meter characteristics.
   1. Operating range.
   2. Accuracy through operating range and consistency of calibration with time.
   3. Resistance to corrosion.
   4. Ability to be disassembled for cleaning if used for foods.

GAS METERS

3.20. Bellows Meters. The familiar household gas meter, Fig. 3.16, consists of two bellows inner connected by valves. As one bellows is being filled from the supply line, the other is emptying

into the service line. Valves shift the direction of flow at the end of the stroke, and the emptied bellows fills from the supply line. The oscillation of the mechanism activates a volumetric indicator.

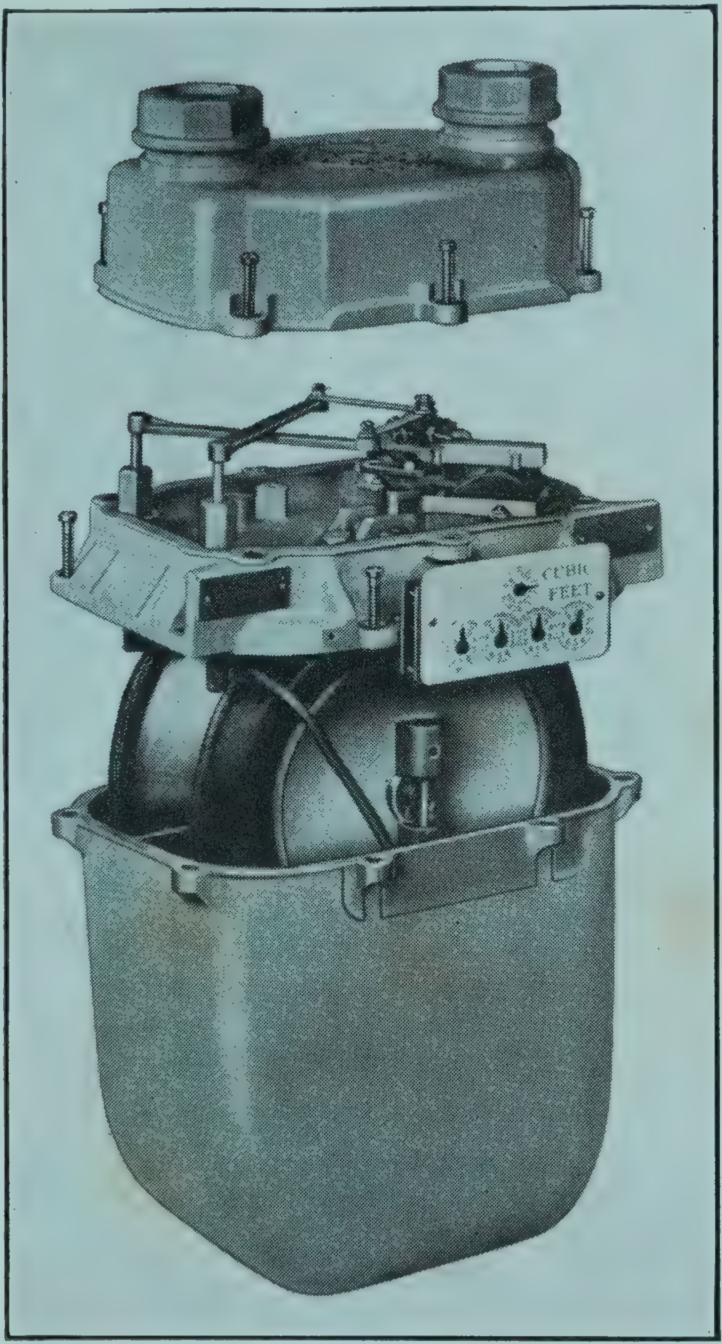


Fig. 3.16. Bellows gas meter. (*Courtesy* Pittsburgh Equitable Meter Division, Rockwell Manufacturing Co.)

The capacity varies from 75 to 10,000 cu ft per hr. Meters are available for operating pressures up to 1000 lb per sq in. The pressure drop through the meter is usually in the order of 0.5 in. of water. The error of a properly operating meter is less than 1.0 per cent.

3.21. Other Gas Meters. Wet gas meters, Fig. 3.17, are very accurate but will not handle high rates of flow or flow that is excessively pulsating. They are used mainly for laboratory or temporary installations. Rotary meters of the vane type are

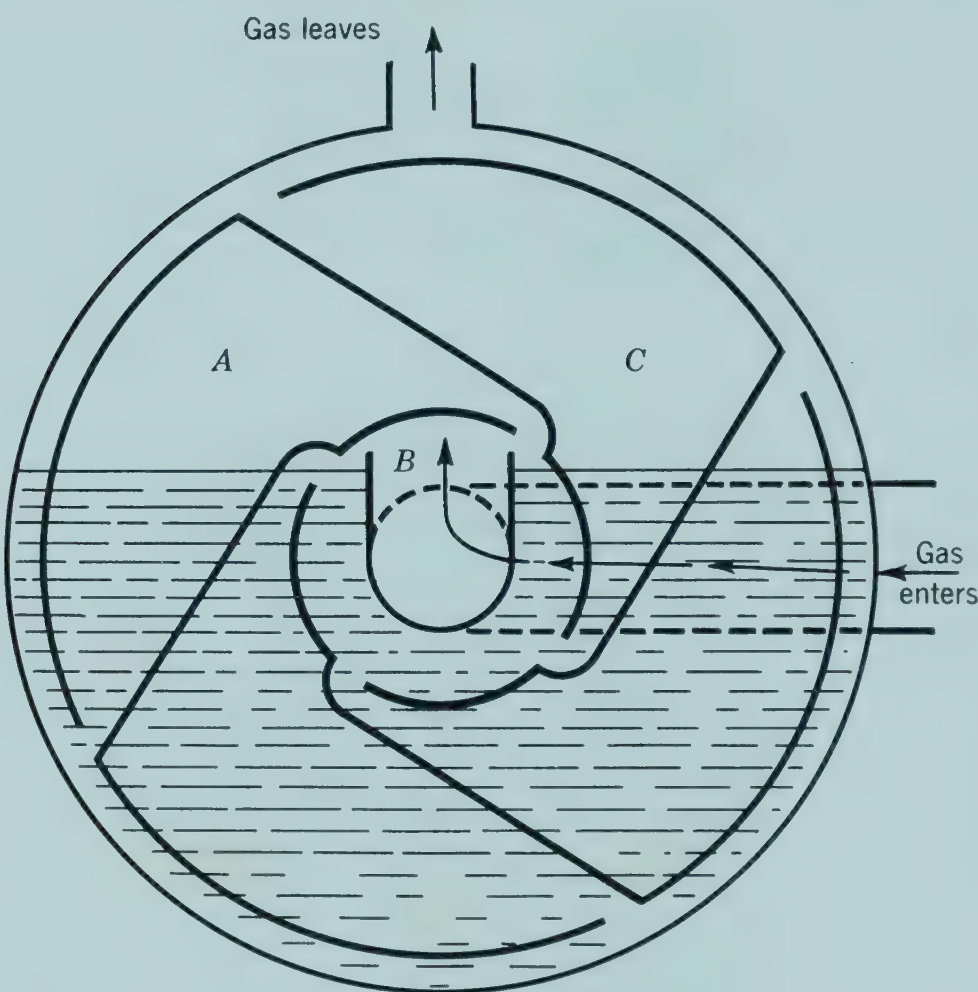


Fig. 3.17. Wet gas meter.

available and perform efficiently within certain velocity brackets. However, they are not as reliable as the other meter types discussed.

LIQUID METERS

Liquid meters are used extensively in many processing operations. In addition to water and petroleum products, they are used for metering brine, acids, alkalies, syrups, milk, fruit and vegetable juices, and other liquids. Metering elements are made of stainless steel, rubber, graphite, brass, bronze, or other materials that will not react with the substance being metered.

3.22. Piston Meters. Piston meters are displacement or volumetric meters since they operate on the basis of and indicate the volume, in cubic feet or gallons, of fluid passed in a certain time.

The meter is similar to the bellows-type gas meter except the bellows are replaced by a double-acting piston. Two double-acting pistons at 90° connected to a crank shaft operate smoother than the single piston. The piston may be fitted with tight-fitted rings or with loosely fitted rings if exact performance is not required. Although piston meters are available for water and other liquids, their main use is for commercial sale of petroleum products where great accuracy is required.

Well-designed piston meters operate with an error of less than 0.2 per cent through the entire operating range. Pressure drop is high because of tight-fitting pistons and may be 5 lb per sq in. at a discharge rate of 100 gal per min, which is the approximate maximum capacity of a meter for use in a 2-in. line.

3.23. Disc and Cylinder Meters. Disc and cylinder meters are recommended for most installations because they are reliable, reasonably accurate, and economical. Disc and cylinder meters are displacement meters, although there is a small clearance between the moving element and the housing which permits a small amount of fluid to be by-passed. However, frictional drag is low and high degree of efficiency is attained. Two types are described.

A hard-rubber disc, frequently called a nutating- or wobble-disc meter is shown in Fig. 3.18. The slit in the disc engages the partition in the metering chamber so that the disc is restrained from rotating. The disc is supported on two hemispherical bearings which permit it to "wobble" but not to rotate. It moves in such a manner that a point on the circumference is continuously in contact with the top of the metering chamber, and an opposite point, with the bottom of the metering chamber, these points moving around the chamber. The axis of the disc moves in such a way that it generates a cone with the vertex at the center of the disc. The partition directs fluid through a channel in the meter which is blocked by the disc. The disc must nutate or wobble to permit the fluid to pass.

Satisfactory performance can be expected for flows greater than 1 gal per min. At lower rates the amount of fluid by-passing the

disc is proportionally high. Rates up to 100 gal per min may be metered on a 1-in. line with an accuracy of 98 per cent or better. Noise and possible decreased efficiency due to wear after prolonged use are the objections to this type of meter. The pressure drop through the meter is small.

Fig. 3.18. Nutating- or wobble-disc meter. (*Courtesy* Pittsburgh Equitable Meter Division, Rockwell Manufacturing Co.)

The oscillating-piston meter shown and described in Fig. 3.19 is similar in principle to the disc meter but superior in performance because of less frictional drag and better balance. The operating range is similar to that of the disc meter, but efficiencies are higher and calibration is more reliable over a period of time. Pressure drops are higher than for the disc meter. It is quiet in operation.

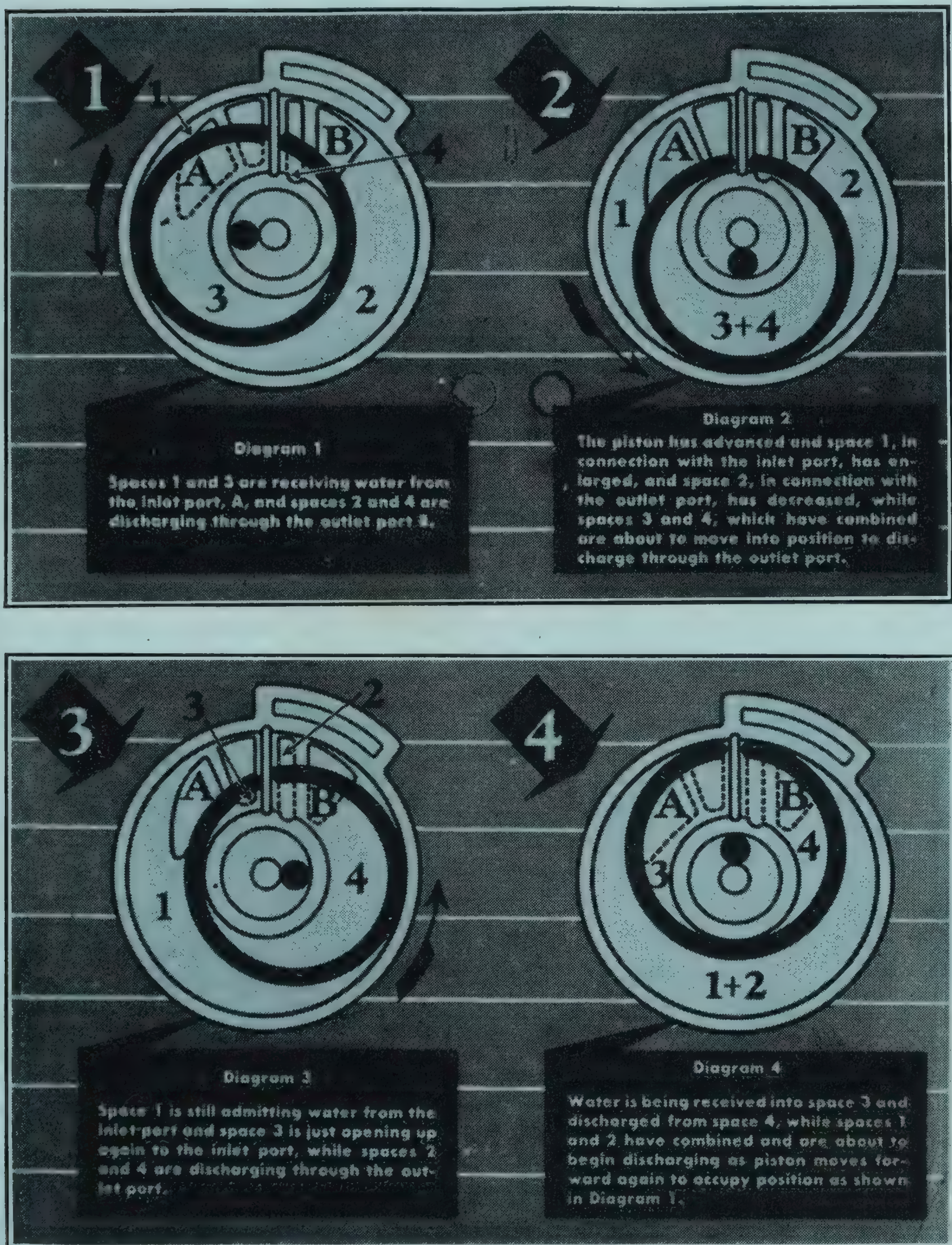


Fig. 3.19. Oscillating-piston meter and its operation. (*Courtesy* Pittsburgh Equitable Meter Division, Rockwell Manufacturing Co.)

3.24. Propeller Meters. Meters that operate from the motion of the fluid rather than the volume flowing are velocity or current, inferential, meters. If restrained from motion, the rate of fluid flow is not altered materially. Vane, propeller, or cup rotors activated by the fluid motion are examples.

They are most useful for continuous high rates of flow such as might be found in washing, irrigation, or general supply lines. They are not efficient at low rates of flow because of bearing friction. Fluids containing solid matter such as dirt and sand can be metered satisfactorily.

A propeller meter is shown in Fig. 3.20.

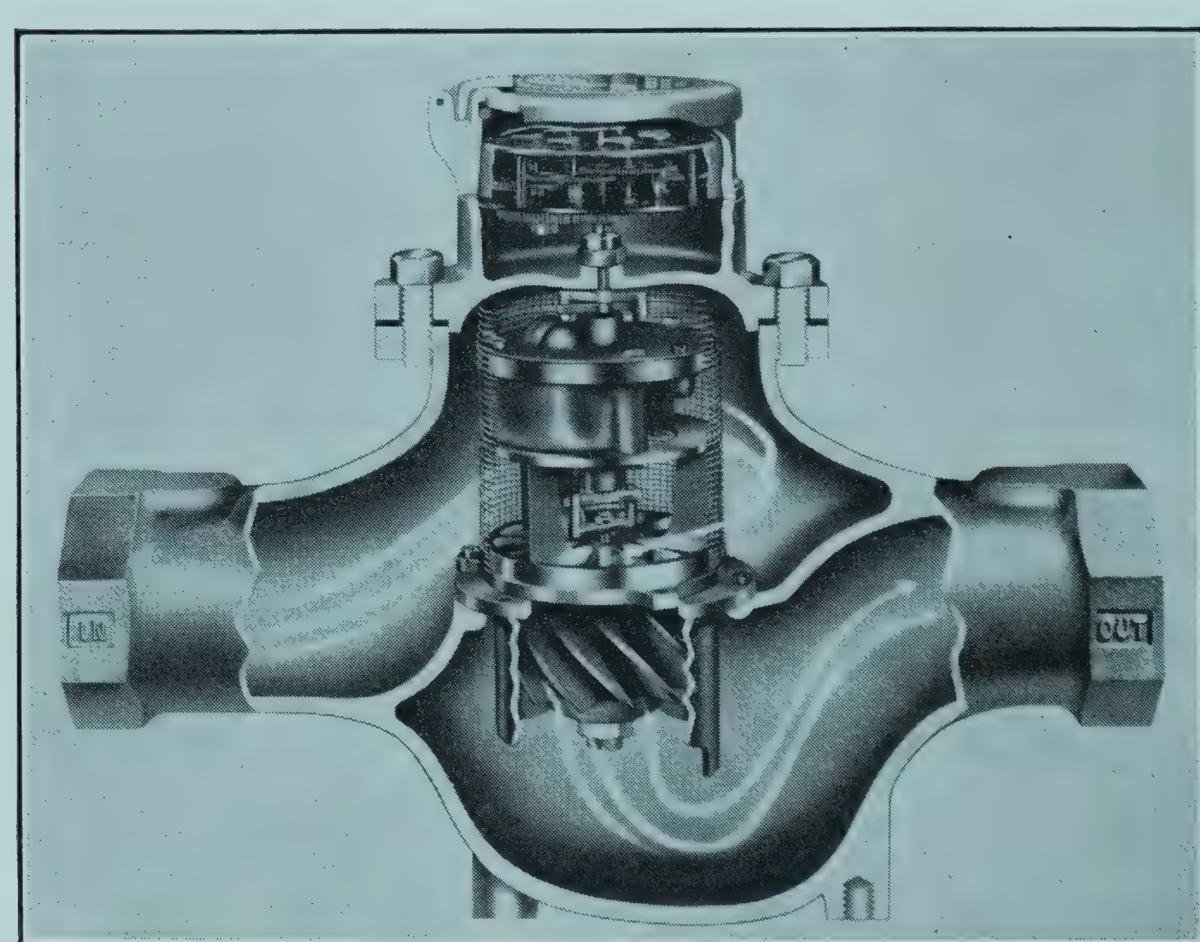


Fig. 3.20. A propeller meter. (*Courtesy* Pittsburgh Equitable Meter Division, Rockwell Manufacturing Co.)

3.25. Rotameters. A commercial rotameter and its schematic elements are shown in Fig. 3.21. The rotor is supported by the upward motion of the fluid, and its position in the tube indicates the rate of flow.

The rotor is stationary when the upward force resulting from flow equals the rotor's weight, or

$$\frac{C\pi d^2\gamma V^2}{8g(144)} = W \tag{3.26}$$

or

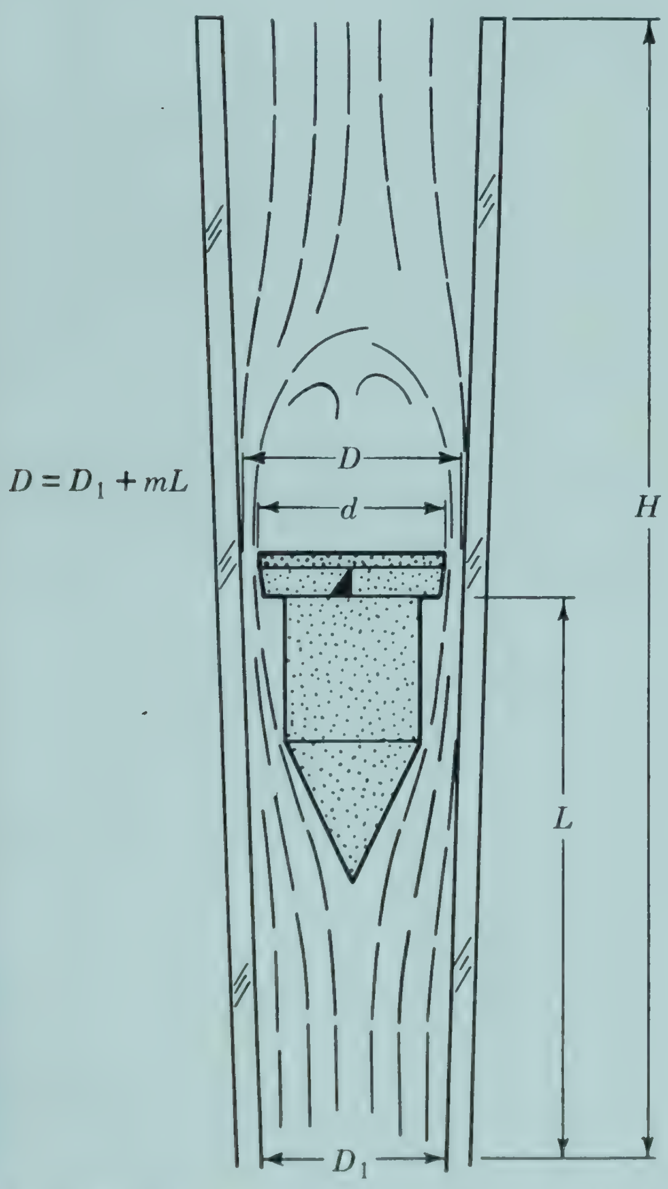
$$V = 153\sqrt{W/d^2\gamma C} \tag{3.27}$$

where V = velocity of fluid at smallest cross section, ft per sec.
d = maximum rotor diameter, in.
γ = specific weight of fluid.
C = rotor drag coefficient; varies with rotor shape and fluid flowing.
W = weight of rotor, lb.

Since the velocity in the above equation must remain constant, the following relationship holds.

$$Q = k(D^2 - d^2) \tag{3.28}$$

in which Q is a quantity rate and the constant k is made up of $\pi/4$ and the required constant velocity value.



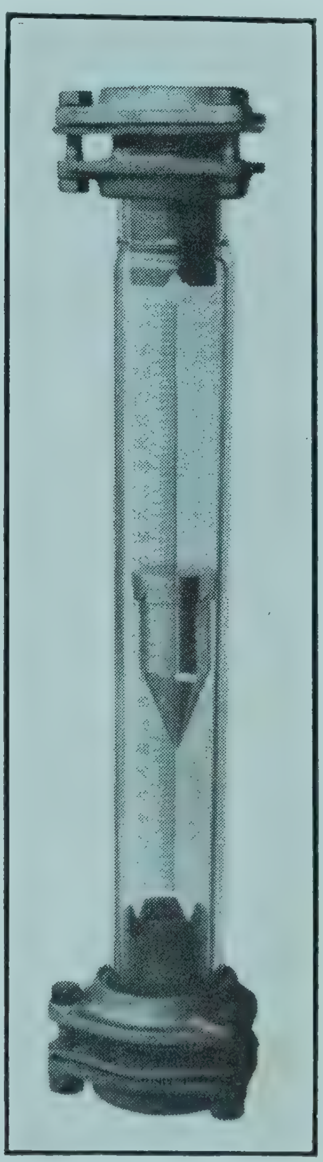


Fig. 3.21. A commercial rotameter with important details shown. (*Courtesy* Schutte and Koerting Co.)

Now, if the difference between D^2 and d^2 is small as compared to d^2, Q may be considered as a linear function of $m(D - d)$ without introducing an error greater than permitted in most engineering work. If a high degree of accuracy is required, the meter can be calibrated through its entire range against a known standard.

Rotameters can be used for liquids or gases. Chemicals, oils, food products, and fluids carrying suspended solid material can be metered. Large or small quantity rates can be handled. The pressure drop through the meter is nominal.

REFERENCES

1. Addison, H. *Hydraulic Measurements.* Second Edition. Wiley. 1946.
2. *Fluid Meters, Their Theory and Applications.* Report of A.S.M.E. Special Research Committee on Fluid Meters, Part I. A.S.M.E. 1937.
3. Folsom, R. G., and Iverson, H. W. Pipe Factors for Quantity Rate Flow Measurements with Pitot Tubes. *A.S.M.E. Paper* 48-A-35. 1948.
4. Amer. Inst. of Physics. *Temperature, Its Measurement and Control in Science and Industry;* Papers presented at a symposium. Pp. 666–672. Rheinhold. 1941.
5. Merriam, K. C., and Spaulding, E. R. Comparative Tests of Pitot-Static Tubes. *National Advisory Committee for Aeronautics. Tech. Note* 546. November, 1935.
6. Ower, E. *The Measurement of Air Flow.* Chapman & Hall. 1933.
7. *Power Test Codes. Information on Instruments and Apparatus,* Part 5, Measurement of Quantity of Materials; Chapter 4, Flow Measurement by Means of Standardized Nozzles and Orifice Plates. A.S.M.E. 1940.
8. Seban, R. A., et al. A Thermal Anemometer for Low Velocity Flow. *Trans. A.S.M.E.* 65:843–846. 1943.
9. Severns, W. H. *Heating, Ventilating and Air Conditioning Fundamentals.* Second Edition. Wiley. 1949.
10. Smith, H. W. Improved Micromanometer. *Ind. Eng. Chem. Anal. Ed.* 8: 151–152. March, 1936.

PROBLEMS

1. A water pressure gauge located 4 ft above a pressure source reads 21 lb per sq in. The connecting tube is full of liquid. What is the actual pressure at the source?
2. The system of problem 1 is initially full of air at atmospheric pressure. The main line was then filled with water, no air being bled from the

gage connecting pipe. What is the pressure in the main line if the gage again indicates 21 lb per sq in.?

3. Air at 70° is being measured by a venturi meter with basic diameters of 14 and 10 in., respectively. Gages attached at points 1 and 2, Fig. 3.11*a*, read 5.75 and 2.10 in. of water. What quantity of air is flowing?
4. The gage on an orifice meter reads 4½ lb per sq in. What is the velocity of water if the inside pipe diameter is 1.25 in. and the sharp-edged orifice is 0.89 in. in diameter? ($C = 0.76$.)
5. A nozzle meter is to be designed to fit into an 18-in. galvanized pipe. Air at room temperature flows through the pipe at velocities varying from 700 to 1400 ft per min. If a 2-in. inclined manometer containing alcohol with a S.G. of 0.89 is to be used with the nozzle, what should be the diameter of the nozzle?
6. A Thomas meter is located in an air duct of 2 sq ft cross-sectional area. The air weighs 0.083 lb per cu ft, and its specific heat is 0.24. Assume a controlled temperature differential of 5° and heater potential of 110 volts. Plot the velocity as abscissa and amperage as ordinate for velocity 0 to 300 ft per min. Assume constant amperage of 3, and plot temperature difference against velocity. Discuss the curves from the standpoint of accuracy of the system.
7. A differential pitot tube located at the center of a cylindrical air tube produces 2.6 in. of water pressure. What is the velocity at the impact end of the pitot tube? The impact gauge reads 2.6, 2.6, 2.5, 2.1, 1.8 when placed at points 1–5 in Fig. 3.10. What is the average velocity in the tube? What factor would be applied to the center reading to indicate a true average velocity? *Note:* In actual practice it would be necessary to check this factor through the entire range of velocities to be encountered, because variation may be expected.
8. A pump is moving soybean oil through a 1-in. (nominal) pipe. The 110-volt motor is using 285 watts. Assuming an over-all pump and motor efficiency of 65 per cent, what is the pumping rate in gallons per minute if the suction and discharge pressures are, respectively, −5 and 23 lb per sq in.?
9. Carbon dioxide is metered into an air conduit which is 12 sq ft in cross-sectional area at a constant rate of 7.0 lb per hour. A sample of the mixture downstream was analyzed by an Orsat apparatus and contained 11½ per cent CO_2 by volume. What is the air velocity?

CHAPTER 4

Pumps

NOMENCLATURE

A = cross-sectional area, sq ft.
C = a proportionality constant.
D = diameter, in.
H = total head, ft of fluid flowing.
H_s = pressure, elevation, and friction head, ft of fluid flowing.
h_f = lift, ft.
h_s = submergence, ft.
hp = horsepower.
N = revolutions per minute.
Q = a quantity of air, cu ft per gal water.
q = cu ft per sec.
V = fluid velocity relative to housing, ft per sec.
V_I = fluid velocity relative to housing tangent to runner, ft per sec.
V_r = fluid velocity relative to vane, ft per sec.
v = peripheral speed of runner, ft per sec.
w = vane width, ft.
y = radial fluid velocity, ft per sec.
γ = specific weight, lb per cu ft.

Pumps are generally considered as devices for elevating or moving liquids. Although this is a satisfactory conception in certain regards, it would be more exact to state that they increase the work head W in the Bernoulli equation (2.7). This restatement implies that the pumping effect upon the fluid might be to elevate the fluid, change its internal pressure, or change its velocity, or a combination of any of these.

The processing engineer is interested in the performance of these devices from the standpoint of their effect upon the three factors indicated above, elevation head, pressure head, and velocity head, and their interrelation when one or more are changed. His chief job will be selection and installation rather than design.

For convenience in discussion, these devices will be classed as follows:

Positive displacement pumps: reciprocating and rotary.
Jet pumps.
Air lifts.
Centrifugal pumps.

4.1. Evaluating Performance. The mechanical efficiency of these devices is the ratio of the work output to the input, the output being the product of the change in the total energy head H times the weight of fluid flowing per minute and the input expressed as foot-pounds per minute. If the input is expressed in horsepower, the equation is,

$$\text{Hydraulic efficiency} = \frac{H \times \text{Lb fluid flowing per min}}{\text{hp} \times 33{,}000}$$

Volumetric efficiency that applies to positive displacement pumps only is the ratio of the volume of fluid moved per unit of time to the piston displacement per unit of time.

PUMP TYPES

The reciprocating or piston type of pump has been treated in previous courses and is well enough known so that a detailed discussion will be omitted. The mechanical efficiency of these pumps may be as high as 80 to 90 per cent, the loss being due in the main to friction. Fluids containing abrasive materials or corrosive fluids cannot be pumped with ordinary reciprocating pumps. They are specially well adapted for high-pressure operation.

4.2. Rotary Pumps. The rotary pumps, Figs. 4.1, 4.2, and 4.3, are positive displacement units and are inexpensive and simple to construct. If constructed and maintained with very close tolerances, the volumetric efficiency is high and high pressures can be produced. A small unit can handle large quantities of fluid because high rotative speeds are possible. Mechanical efficiencies may be 90 per cent or more under the best conditions.

Gear pumps are well suited for many processing operations since they are positive acting, provide a continuous smooth flow of material, do not whip the material when pumped, and are easily disassembled for cleaning. They are specially suitable for viscous substances such as ice-cream mix, molasses, and oils.

Gear pumps perform best when pumping fluids with some lubricating properties. However, external gears are frequently pro-

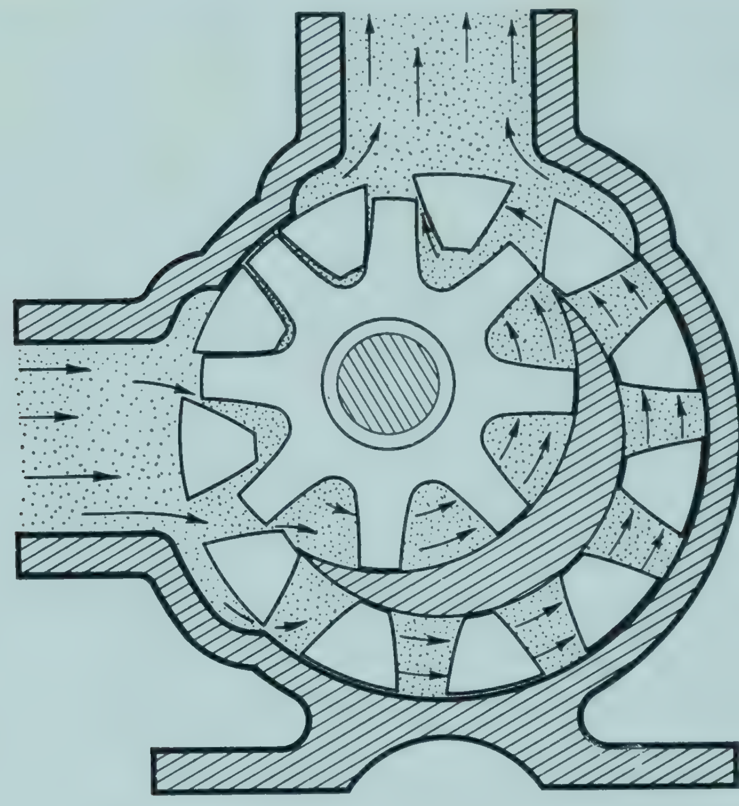


Fig. 4.1. Internal-gear rotary pump.

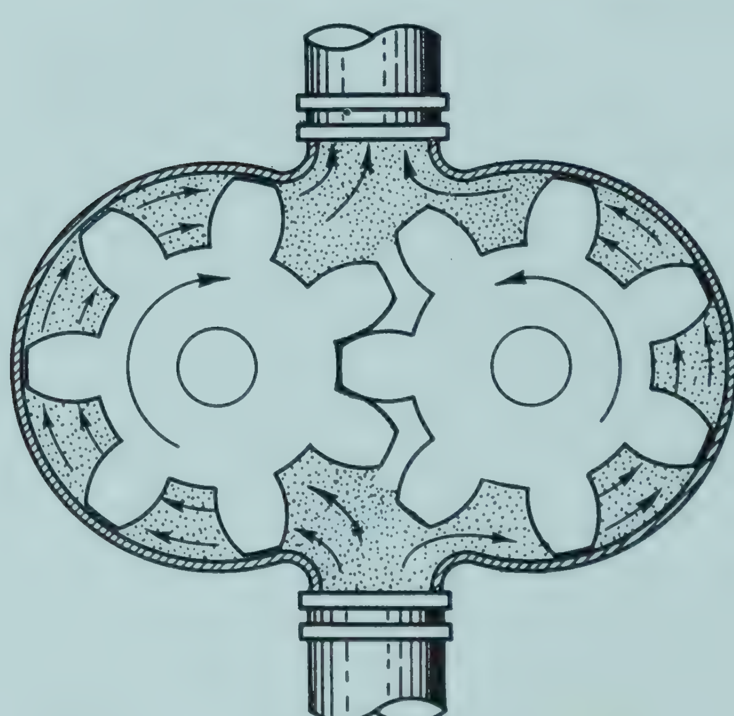


Fig. 4.2. External-gear rotary pump.

vided to maintain pumping-gear alignment. This feature permits nonlubricating fluids to be pumped with a minimum of wear to the pump. For high-pressure operation, close tolerances must be

provided and maintained. Satisfactory performance persists only if close tolerances are maintained.

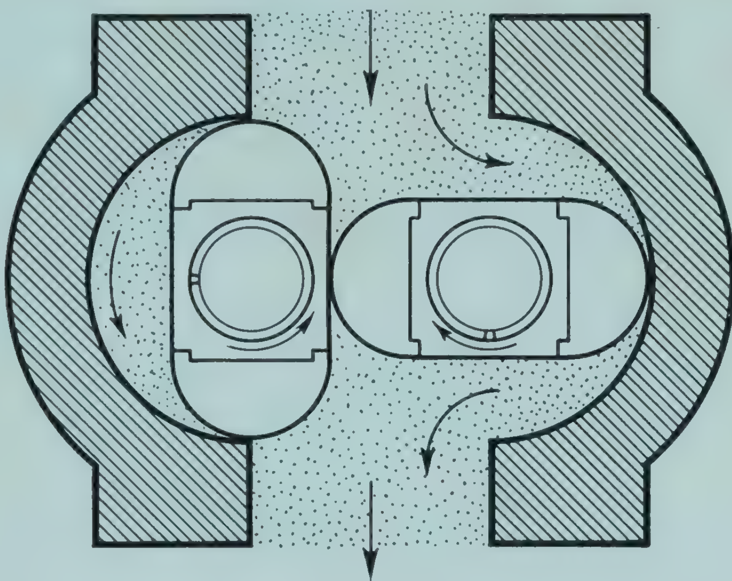


Fig. 4.3. Lobe pump, used for both gases and liquids.

Vane pumps are characterized by sliding vanes such as shown in Fig. 4.4, cylindrical seals in slots, and hinged lobes or other details for maintaining a seal in the pump. This pump, although

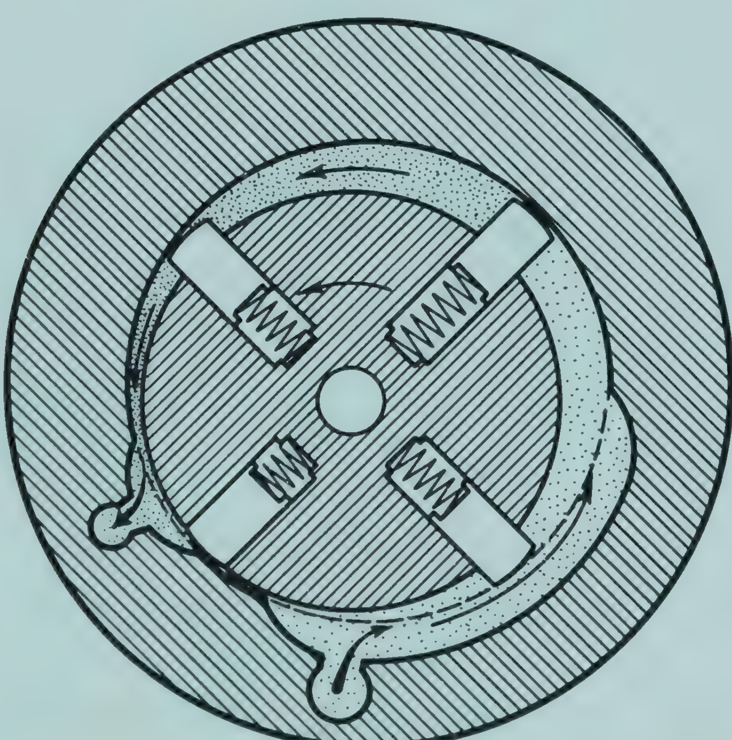


Fig. 4.4. Vane pump, used for both gases and liquids.

more subject to wear than other rotary pumps, can develop a high pressure because of a better seal between the rotor and the housing. Its chief use in the processing field is evacuating, particu-

larly for dry vacuum pump work such as that done by milking machines and vacuum pans.

4.3. Jet Pumps. The jet pump shown elementarily in Fig. 4.5 operates on the velocity energy of a jet of fluid. Water (or

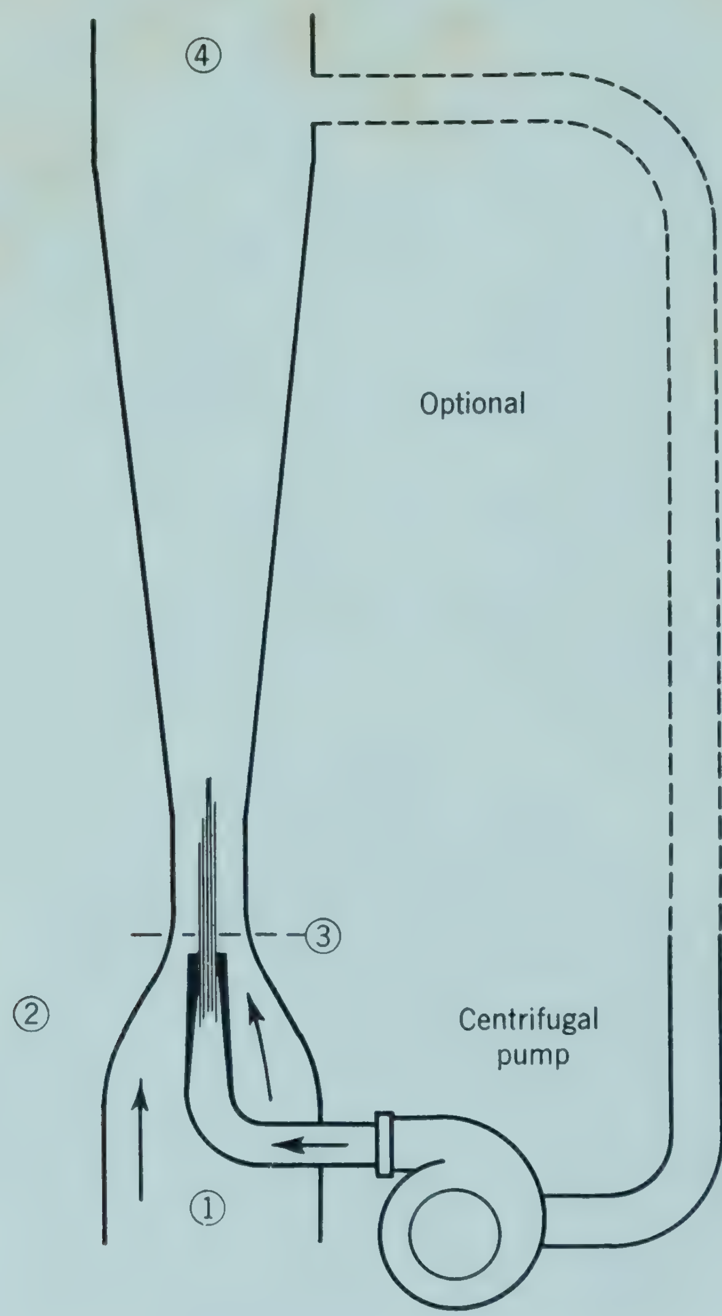


Fig. 4.5. Schematic elements of a jet pump.

other fluid, either compressible or incompressible) is forced through a jet or nozzle of such dimensions that all or nearly all the energy involved is converted into velocity energy. This energy, which is directional, is applied to the fluid to be moved. The jet is produced by recirculating a portion of the liquid or gas in those cases where the material will not damage the pump.

Jet pumps are frequently used for pumping sumps or processing residues that contain solid matter or chemically active materials that would not pass through a mechanical pump satisfactorily. For example, if the material is a gas, water or air is provided from an external source to supply the jet energy. The diluted mixture is discarded.

Jet-pump theory as presented by Gosline and O'Brien [4] is rational and straightforward but too involved to include in this

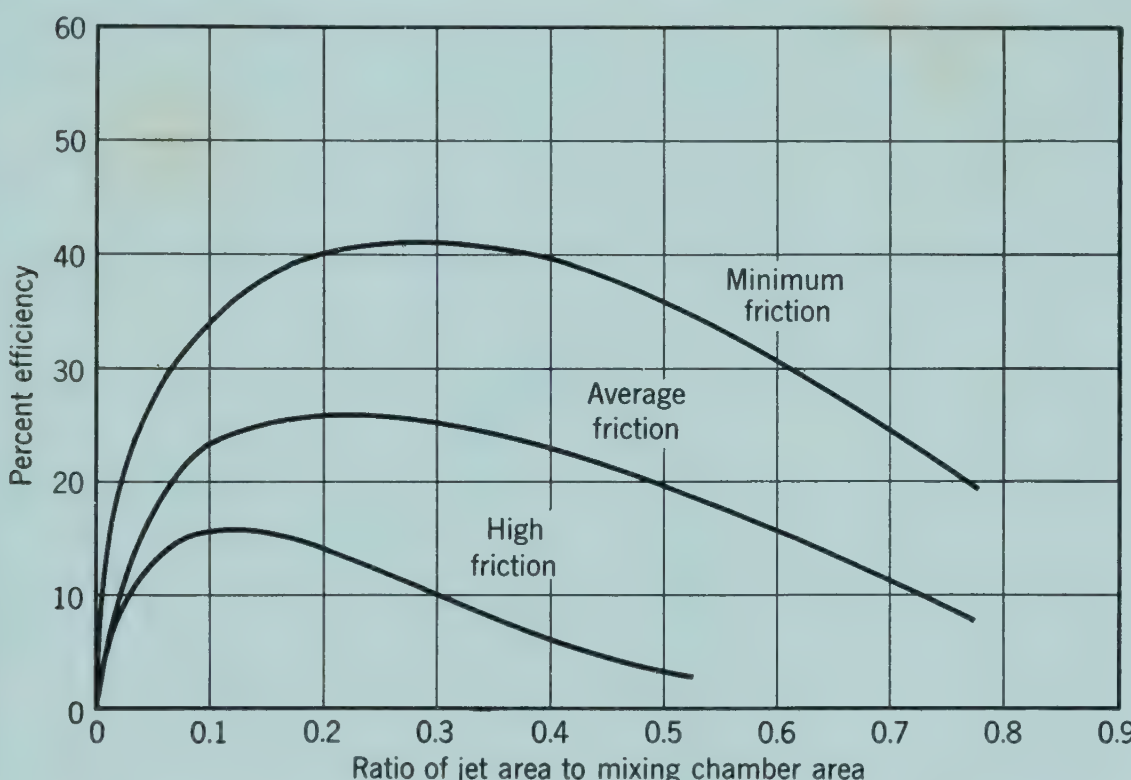


Fig. 4.6. Jet-pump efficiency related to the relative jet area and friction in the jet unit.

text. However, the basic consideration for a single fluid is the conservation of energy, which may be expressed thus:

$$H_4A_4V_4 - H_1A_1V_1 + F = H_2A_2V_2 \tag{4.1}$$

H, which is the total hydraulic or Bernoulli head, can be made up of any combination of elevation, pressure, velocity, and resistance heads. However, H_2, the power head, must be predominately velocity head in order to transfer the energy to the fluid entering at 1 in Fig. 4.5. The power loss due to friction and turbulence F, which can be calculated, represents the energy loss resulting from mixing the fluids from regions 1 and 2 in region 3. The efficiency may be expressed thus:

$$\frac{H_4A_4V_4 - H_1A_1V_1}{H_2A_2V_2} \tag{4.2}$$

The efficiency is closely related to the ratio of the nozzle area to the mixing-cylinder area and the friction in the system. The relationships of these factors as reported by Gosline and O'Brien are shown in Fig. 4.6. These are theoretical curves that have been substantiated by observation.

In spite of its low efficiency, the simplicity of the jet pump, its freedom from moving parts, its ability to pump materials of sludge consistency, and its low initial cost fit it for use in situations where other pumping devices would be impractical.

4.4. Air Lift. Another convenient device for elevating liquids is the air lift shown in Fig. 4.7. Air is delivered at the bottom of the lift pipe and mixes with the liquid. The air-liquid mixture,

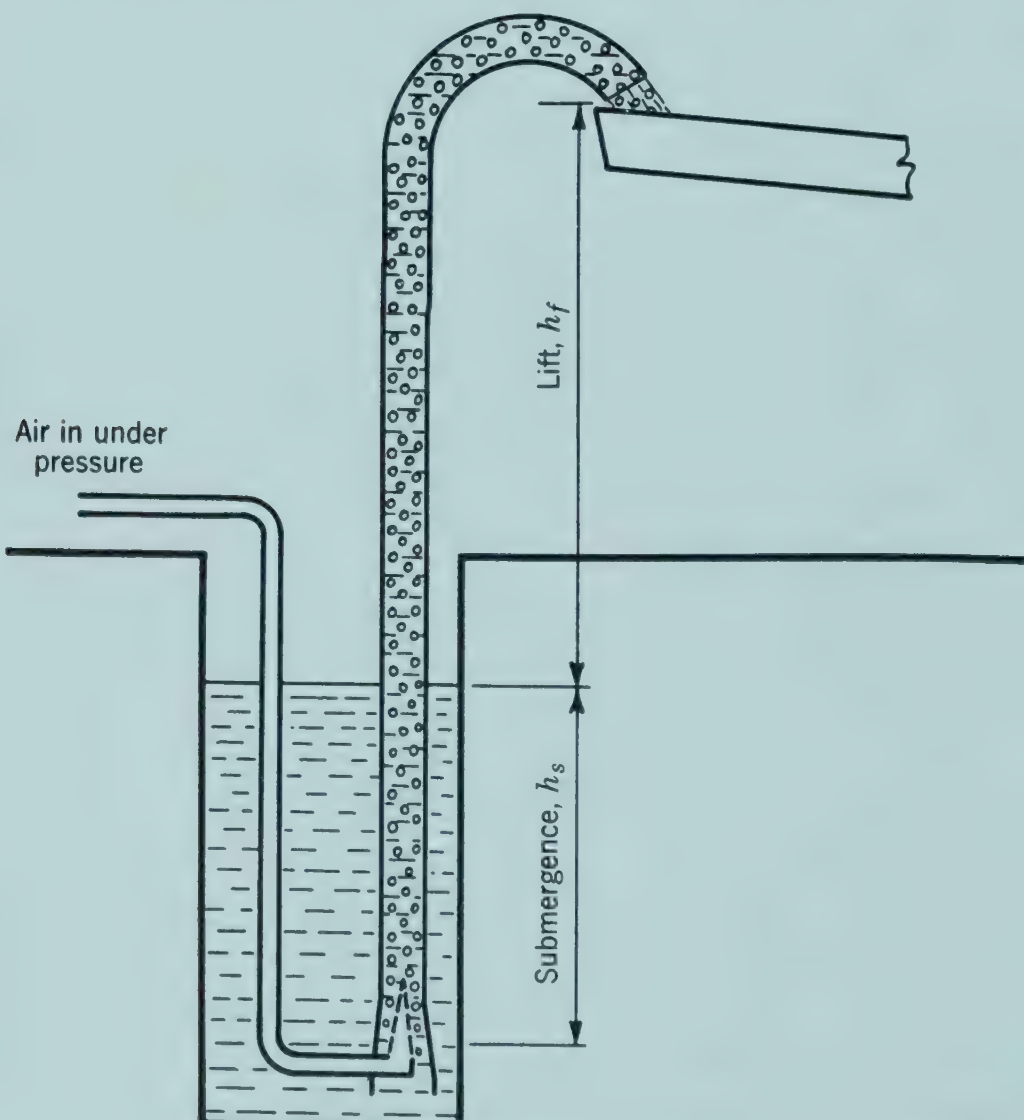


Fig. 4.7. The air lift.

being of less specific weight than the liquid, rises in the pipe and is discharged at a point above the level of the liquid.

An empirical formula has been developed by the Ingersoll-Rand Company for design where water is the fluid being lifted. It is

$$Q = 0.8 \frac{h_f}{C \log [(h_s + 34)/34]} \tag{4.3}$$

where Q = cu ft free air required to lift 1 gal of water.

h_f = lift, ft.

h_s = submergence, distance from surface of water to point of air introduction, ft.

C = a constant (note table).

Lift h_f, ft	C
10–60	245
61–200	233
201–500	216
501–650	185
651–750	156

The submergence, expressed as the ratio $h_f/(h_s + h_f)$, should vary from 0.66 for a lift of 20 ft to 0.41 for a lift of 500 ft. Foot-piece design is important since the size and distribution of air bubbles materially affect the efficiency. The air compressor must be able to supply air at a pressure head equal to h_s and in a quantity specified by equation 4.3.

This lifting device * can be employed for fluids containing foreign materials and for corrosive liquids. From 20 to 40 per cent of the energy used to compress the air is effective in elevating the liquid.

4.5. Centrifugal Pump. The centrifugal pump is widely used for pumping water, milk, lubricants, chemical solutions, materials being processed, etc. Its popularity is due to relative simplicity, mechanical efficiencies as high as 90 per cent under favorable conditions, and ability to handle fluids containing solids in suspension. Centrifugal pumps can be designed for high-pressure operation where necessary. Because of simplicity and ease of disassembling, which facilitate cleaning, washing, and sterilizing, they are satisfactory for food products.

* This device is not a pump in the strictest sense since it cannot alter the pressure or velocity heads in the Bernoulli equation in any practical degree.

The basic principles of design that also apply to fans and blowers are important to an understanding of performance and proper selection. Considering Fig. 4.8 we assume that (1) the blade thickness is negligible; (2) friction losses are negligible; and (3) the peripheral velocity at the inlet is zero.

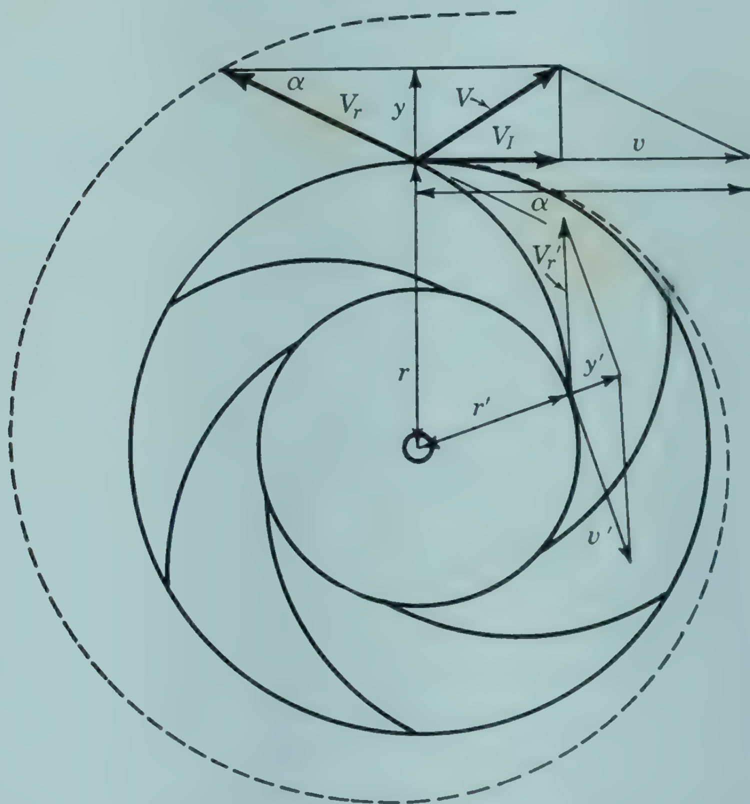


Fig. 4.8. Velocity vectors for a centrifugal-pump runner.

The angular momentum of the water immediately after leaving the impeller is

$$(W/g)V_I r \qquad (4.4)$$

which may also be considered the torque exerted on and by the impeller. The work done per second by the impeller upon the water is

$$\text{Torque} \times \text{Angular velocity} = (W/g)V_I r \times v/r = (W/g)V_I v \qquad (4.5)$$

The work input per second (or other unit of time) per pound of water flowing is

$$(V_I v)/g \qquad (4.6)$$

If the input is referred to a reference plane through the pump, the output from the standpoint of Bernoulli's theorem is

$$H + (V^2/2g) \tag{4.7}$$

in which H_s may be either pressure, friction, or elevation head or a combination of them. Equating equations 4.6 and 4.7, the theoretical mechanical-energy balance is

$$(V_I v/g) = H_s + (V^2/2g) \tag{4.8}$$

or

$$H_s = (V_I v/g) - V^2/2g \tag{4.9}$$

Now $V^2 = V_I{}^2 + y^2$ and $V_I = v - y \cot \alpha$, from which

$$V^2 = (v - y \cot \alpha)^2 + y^2 \tag{4.10}$$

Substituting for V^2 and V_I in equation 4.9 gives

$$H = \frac{2v(v - y \cot \alpha) - (v - y \cot \alpha)^2 + y^2}{2g} \tag{4.11}$$

which when solved becomes

$$H_s = \frac{v^2 - y^2 \csc^2 \alpha}{2g} \tag{4.12}$$

or

$$\frac{v^2}{2g} = H_s + \frac{y^2 \csc^2 \alpha}{2g} \tag{4.13}$$

Consider the meaning of this equation from the standpoint of pump design and selection. The first term $v^2/2g$ is directly related to the speed of the runner. H_s is the static friction and/or elevation head against which the pump is operated. The rate of discharge equals $2\pi rwy$ and is represented by y. The shape and depth of the vane are the two basic design features that affect performance. Assuming no friction loss in the runner and a constant speed, that is, v constant, note the following very significant features.

1. If the operating head H is increased, the rate of discharge represented here by y, decreases. This response is a characteristic feature of centrifugal pumps.

2. The discharge velocity y may be varied by the runner design. Since the mass rate must be the same at the inner and outer

peripheries, the capacity per revolution $= 2\pi rwy = 2\pi r'w'y'$, in which the prime values are at the inner periphery. The velocity y decreases as r' decreases and as w' decreases. y is not constant, because of resistance loads. Therefore, pumps with deep vanes and narrow inner peripheries produce high static heads with low discharge rates.

3. An increase in the vane pitch angle permits a decrease in the revolutions per minute required for a certain discharge and decreases the maximum head under which satisfactory performance can be expected.

4. The total theoretical head at complete shut-off is equal to the square of the peripheral speed divided by $2g$, equation 4.12 with y equal to 0. Actual shut-off heads sometimes exceed the theoretical. This occurrence is believed to be due to fluid circulation within the pump casing.

Pumps are used to move a quantity of fluid against a resistance which may be attributable to elevation or friction of conduits, nozzles, and other fittings. Therefore, the velocity head produced by a pump should be converted to static head H. This conversion is attempted by gradual reduction of the velocity in one of two ways.

Diffuser or guide vanes may conduct the fluid away from the impeller and gradually lower its velocity by increasing the conduit area. The reduction in velocity effects an increase in pressure head as a result of the operation of the Bernoulli equation. The vanes are so bent that the water is turned gradually and is finally discharged into a manifold.

A second method, which is simpler and less expensive, is the volute manifold or casing outlined by the dotted line in Fig. 4.8. The casing is so designed that the average velocity is constant at all cross sections and is approximately $V/2$ in Fig. 4.8. When properly designed, each fluid element is gradually turned toward the discharge outlet so that turbulence losses are at a minimum.

Further reduction in velocity effect may be had by gradually expanding the diameter of the discharge pipe.

4.6. Performance, Testing, and Rating. The American Society of Mechanical Engineers and the Hydraulic Institute[5,7] have developed standard methods for testing centrifugal and rotary pumps. These test code series should be studied if formal

tests are to be made or performance data subjected to a critical analysis.

Tests are made by operating the pump at a constant speed and varying the capacity by throttling the outlet. The total head, the velocity head, and perhaps the static head are plotted against the rate of discharge. The horsepower input and efficiency are also determined and plotted against the discharge rate. The

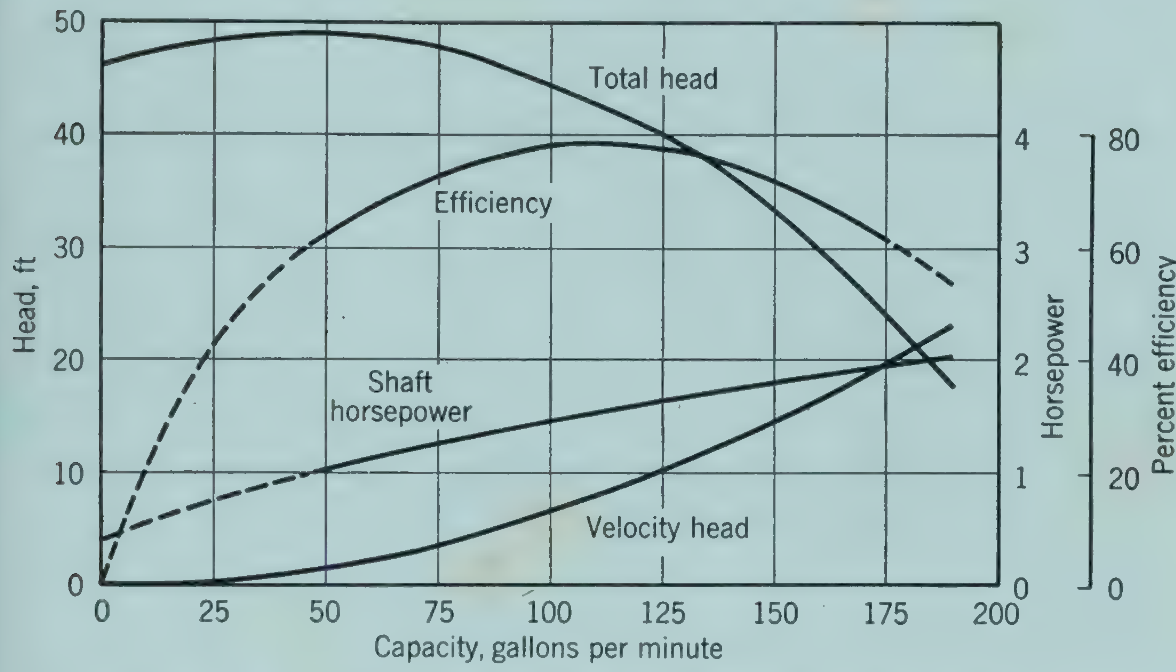


Fig. 4.9. A representative standard centrifugal-pump test plot.

horsepower input to the pump shaft is determined by any of the accepted procedures. Calibrated electric motors are included. The efficiency is expressed as the ratio of the fluid horsepower to the shaft horsepower where,

$$\text{Fluid horsepower} = q\gamma H/550 \tag{4.14}$$

where q = cu ft fluid per sec.
γ = fluid specific weight, lb per cu ft.
H = total head, ft.

A complete performance study would include a series of tests made at different pump speeds.

A representative pump test plot is shown in Fig. 4.9. The pump is a $2 \times 2\frac{1}{2} \times 7$ * pump direct connected to an electric motor that operates at 1760 rpm. Note that some power is required at the no-discharge position.

* Outlet 2 in. in diameter, inlet 2½ in., and runner 7 in. in diameter.

A specific job would require a pump with a specified capacity at a specified head. Selection would be made from a series of test plots for different-size pumps. As a general rule it is advisable to select a pump so constructed that the point of performance will fall to the right of the maximum efficiency point on the test sheet. Then, if the operating head increases after being placed in operation, the efficiency will not be affected and

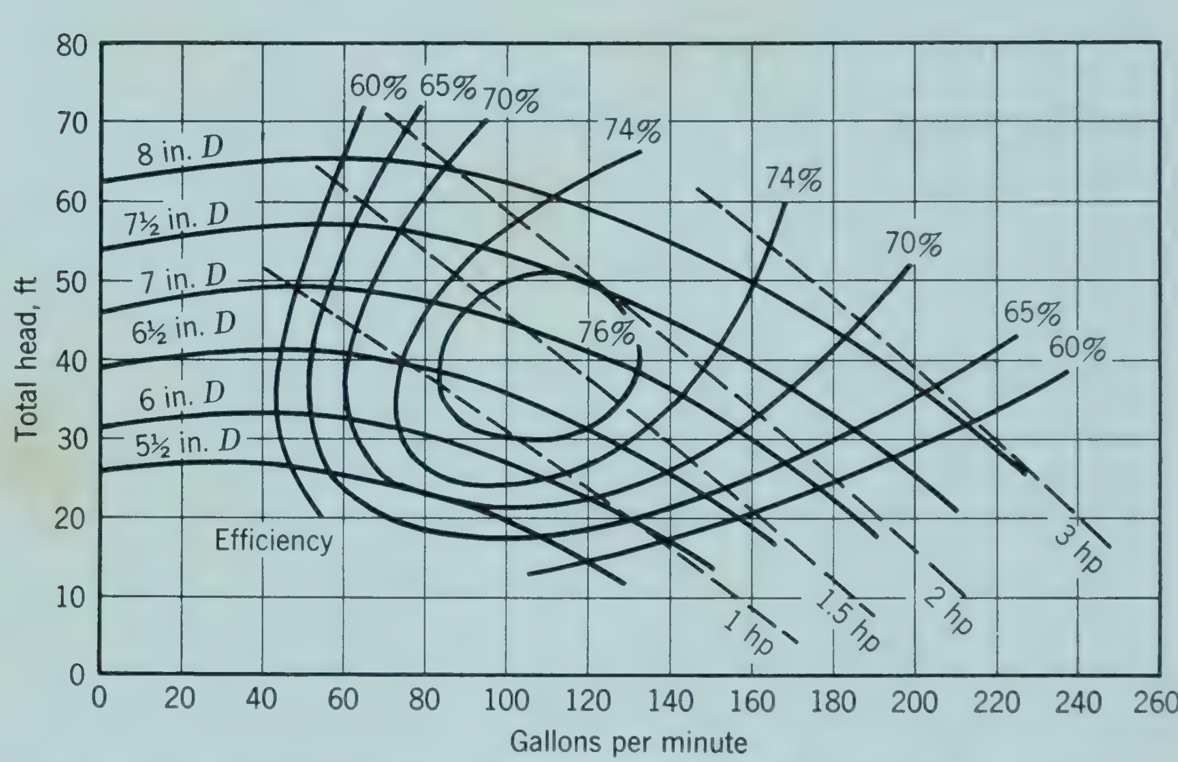


Fig. 4.10. Performance data for a $2 \times 2 \times 8$ in., 1760 rpm centrifugal pump with impellers ranging in diameter (D) from 5 to 8 in.

the capacity will not be lowered significantly. For example, if the total head is increased from 35 to 40 ft because of added lift or restriction in the line, the capacity will drop from 145 to 125 gal per min, 14 per cent, and the efficiency will *increase* from 74 to 78 per cent. On the other hand, a similar total head increase from 42½ to 47½ ft would reduce the capacity from 112½ to 80 ft, a reduction of 29 per cent as compared to 14 per cent above. The efficiency would *decrease* from 79 to 74 per cent.

Performance data are sometimes presented as shown in Fig. 4.10. This isoefficiency plot shows the performance of a pump with a number of different runners. Similar plots with a single runner operating at different speeds are common. Selection should usually be made to the right and below the point of highest efficiency for reasons as noted above.

Commercial performance data are usually made available in tabular form. The performance tables are composed from performance curves such as those discussed.

4.7. Regenerative Turbine Pump. The regenerative turbine pump shown in Fig. 4.11 is a simple rotating pump with certain characteristics superior to the centrifugal pump.

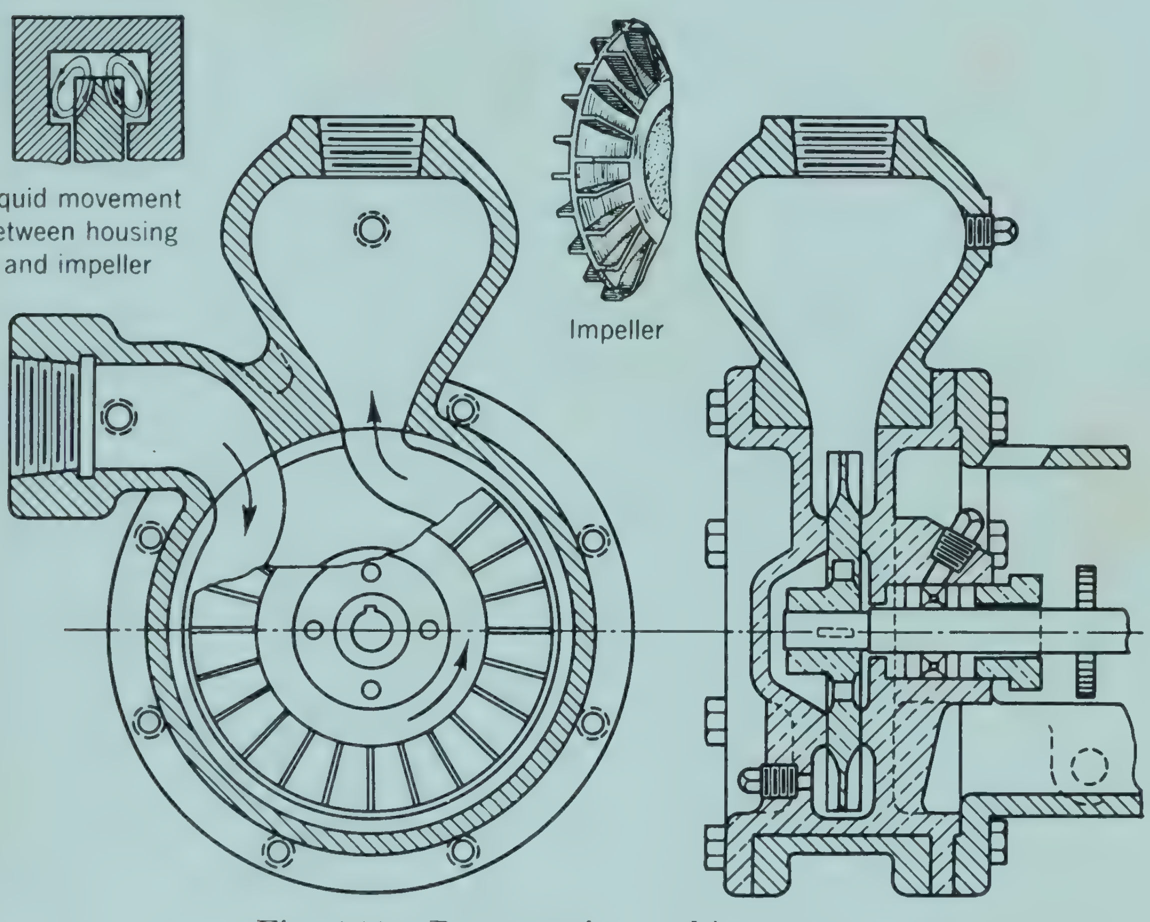


Fig. 4.11. Regenerative turbine pump.

The impeller operates in a closely machined channel. The fluid is moved through the channel by small blades that are machined in the rim of the impeller. Energy is supplied to the liquid by circulation between the impeller and the housing as shown in Fig. 4.11. Each time the fluid circulates, energy is supplied to it which raises the operating head. This recirculation is somewhat comparable in performance and effect to multistaging of centrifugal pumps, but it results from a single impeller. For a low discharge head the velocity is high and the number of circulations are at a minimum. As the discharge is throttled, the rate of discharge is reduced and circulation is increased, thus increasing the operating head.

A performance curve of a regenerative turbine pump is shown in Fig. 4.12. In comparing this type of pump with the centrifugal pump three distinguishing features are noted.

1. A higher operating head can be developed by a single-stage pump with the same impeller diameter.

2. The required power decreases as the capacity increases (or operating head decreases). This is inverse to the centrifugal pump.

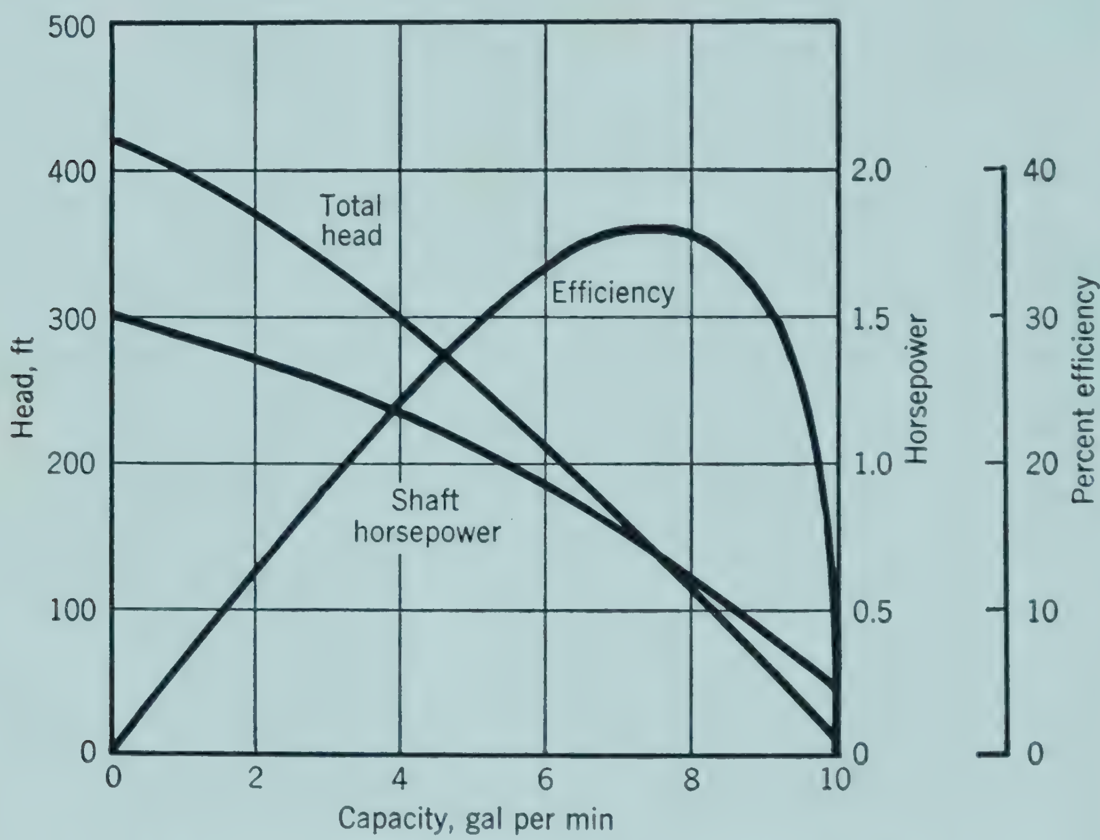


Fig. 4.12. Performance data for a regenerative turbine pump.

3. The efficiency is lower on large capacity pumps, but it is equal or higher on capacities up to 35 gal per min.

This type of pump is made in small capacity sizes for operation under high working heads. It is specially suited for deep-well domestic water systems, especially deep-well jet pumps, boiler-feed applications, high-pressure washing and spraying, and other similar services. The low efficiency should not be considered as a detrimental factor since high working heads can be developed without multistaging.

4.8. Performance of Rotary Pumps. A rotary pump performance curve is presented in Fig. 4.13. This type of pump, which is considered a positive displacement pump, was discussed

in Sect. 4.2. The performance curve is presented here in order to permit the student to compare rotary pump performance with the performance of the other pumps discussed.

The pump represented here is an internal gear pump designed for high-pressure operation. At no working head, the capacity is 25 gal per min and 0.65 hp is required. Pump resistance accounts

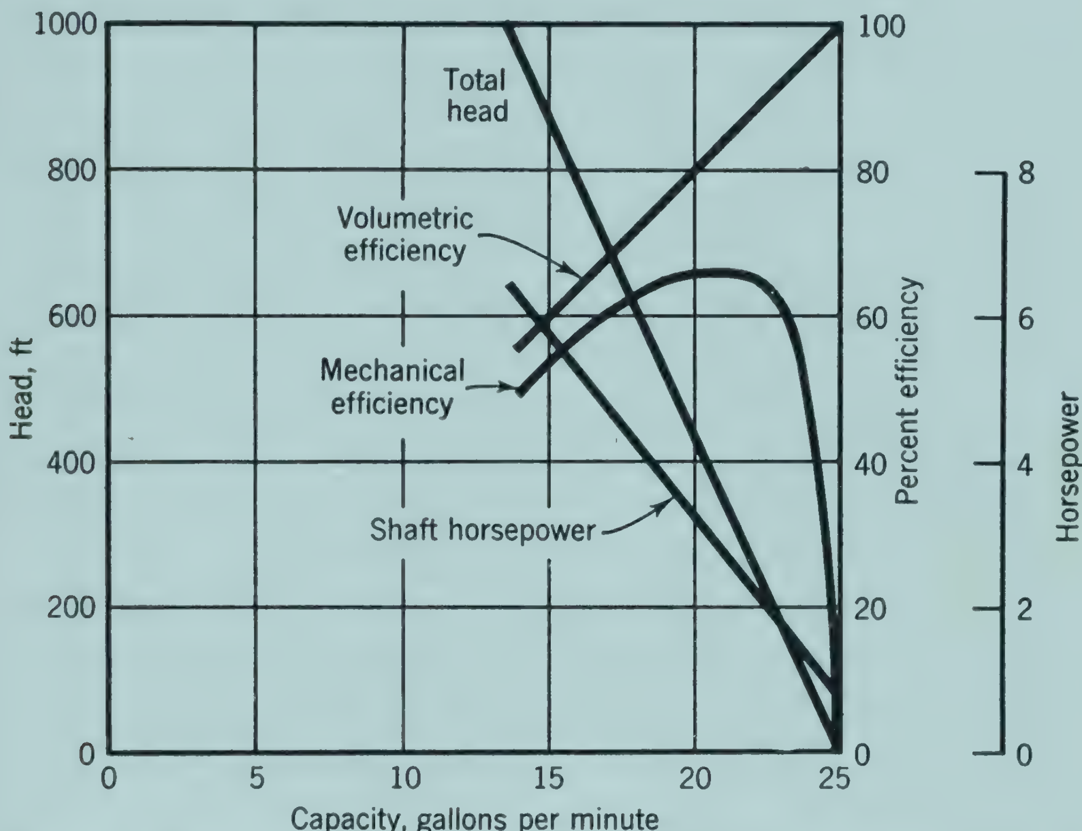


Fig. 4.13. Performance of a gear pump. Fluid weighs 7.75 lb per gal.

for the power requirement at this condition. As the working head is increased, the power increases almost proportionally. If there were no back leakage between the gears and housing, the volumetric efficiency would be 100 per cent and the head and power curves would be vertical. The decrease in volumetric efficiency in this example should not be considered as discrediting this particular pump since the operating head is so high.

4.9. Multistaging. The velocity, amount of lift, and static head may be increased by using two or more single-unit or single-stage pumps in series. Since the total head produced by a pump, equation 4.13, is the difference between the intake and discharge heads and no reference is made to the absolute value of either, it is easy to understand that the discharge head of one pump could

be the intake of a second pump. This condition holds only if the capacity of both pumps is the same.

Multistage pumps are usually designed with the impellers on the same shaft and with straightening vanes between each impeller. Design procedures are not relevant enough to our study to include here.

4.10. Centrifugal Pump Laws. The performance of pumps, fans, and blowers conforms to certain mathematical laws that have widespread application. These laws may be used to extend the performance data for a specific pump (or fan) to a geometrically similar pump of a different size or operating at a different speed. Geometric similarity implies that all comparable dimensions of the pumps being considered are proportional, that the friction factors do not change appreciably over the Reynolds-number range considered, and that the efficiency is constant. The internal resistance of the pump is not strictly proportional to the total head, and there is some variation in Reynolds number. These factors complicate the critical application of the laws, but the error resulting is of no practical consequence.

These laws apply to the pump only and not to the system to which it is attached. Also they apply to a specific point on the performance curve. Extrapolation of a specific point on a curve will give a new point similarly located on a new set of curves. These laws follow.

I. For a specific pump with speed varying.

1. The capacity varies directly as the speed (N equals revolutions per minute). This is true since y is proportional to v in Fig. 4.8 and v is proportional to speed.

$$N_1/N_2 = q_1/q_2 \tag{4.15}$$

2. *The total pressure head varies as the square of the speed.* This follows because of the basic relation between speed and pressure.

$$N_1^2/N_2^2 = H_1/H_2 \tag{4.16}$$

3. *The power required varies as the cube of the speed.* Since power is a product of the rate of mass motion and force or pressure, hp $\propto qH$. And since $q \propto N$ and $H \propto N^2$, it follows that

$$N_1^3/N_2^3 = \mathrm{hp}_1/\mathrm{hp}_2 \tag{4.17}$$

Example. A pump, operating at 1760 rpm, delivering 125 gal per min at 40 ft of head, and requiring 1.63 horsepower, is speeded up to 2100 rpm. What are the new operating conditions? The capacity, equation 4.15, is

$$1760/2100 = 125/q_2$$

$$q_2 = 149 \text{ gal per min}$$

The total head produced, equation 4.16, is

$$1760^2/2100^2 = 40/H_2$$

$$H_2 = 56 \text{ ft}$$

The power required, equation 4.17, is

$$1760^3/2100^3 = 1.63/\text{hp}_2$$

$$\text{hp}_2 = 2.76$$

(Note that, because of similarity, the new operating position is at the same relative position regarding efficiency as the initial operating point.)

II. For a number of geometrically similar pumps with speed constant and diameter varying.

1. *The capacity varies as the cube of the diameter.* The capacity is a direct function of periphery speed and periphery area. Since the periphery speed varies directly as the diameter and the area as the square of the diameter, it follows that

$$D_1{}^3/D_2{}^3 = q_1/q_2 \tag{4.18}$$

2. *The head varies as the square of the diameter.* The pressure varies as the square of the velocity. The velocity of the fluid leaving the impeller varies as the diameter of the propeller, consequently,

$$D_1{}^2/D_2{}^2 = H_1/H_2 \tag{4.19}$$

3. *The power varies as the fifth power of the diameter.* Since power is the product of quantity discharge and head, equations 4.16 and 4.17 combined in product show that

$$D_1{}^5/D_2{}^5 = \text{hp}_1/\text{hp}_2 \tag{4.20}$$

Example. A pump with a 7-in. runner delivers 125 gal per min against a 40-ft head and requires 1.63 hp. If the speed is maintained constant and the runner diameter increased to 7.42 in. (because of similarity, all

linear dimensions would be increased proportionally), what are the new operating conditions?

The capacity, equation 4.18, is

$$7^3/7.42^3 = 125/q_2$$

$$q_2 = 149 \text{ gal per min}$$

The total head produced, equation 4.19, is

$$7^2/7.42^2 = 40/H_2$$

$$H_2 = 45 \text{ ft}$$

The power required, equation 4.20, is

$$7^5/7.42^5 = 1.63/\text{hp}_2$$

$$\text{hp}_2 = 2.18$$

(As in the previous example the new operating point bears the same relation to the new efficiency curve as the initial point bears to the initial efficiency curve.)

Note that the two examples were taken from the performance data of Fig. 4.9. The capacity or rate of discharge was raised from 125 to 149 gal per min first by increasing the speed of the runner from 1760 to 2100 rpm and then by increasing the diameter of the runner. Although the rate of discharge was the same in both instances, the total operating head resulting from the speed increase was 56 ft, but that resulting from runner-diameter increase was only 45 ft. Comparable variations in power requirement are noted.

These variations would lead one to believe that any head between 45 and 56 ft could be produced by adjusting both speed and diameter. Fortunately, a procedure is available for extending performance data to any desired condition from any operating point on a performance chart.

The development of this procedure follows:

The pump laws may be combined so that the speed and diameter effect appear in the same equation, thus:

$$q_1/q_2 = (N_1/N_2)(D_1^3/D_2^3) \tag{4.21}$$

$$H_1/H_2 = (N_1^2/N_2^2)(D_1^2/D_2^2) \tag{4.22}$$

$$\text{hp}_1/\text{hp}_2 = (N_1^3/N_2^3)(D_1^5/D_2^5) \tag{4.23}$$

By solving equations 4.21 and 4.22 simultaneously, the following generalized expressions are secured.

$$D_2 = D_1(H_1^{1/4}/q_1^{1/2})(q_1^{1/2}/H_2^{1/4}) \tag{4.24}$$

$$N_2 = N_1(q_1^{1/2}/H_1^{3/4})(H_2^{3/4}/q_2^{1/2}) \tag{4.25}$$

The application of these most important expressions and of equation 4.23 are demonstrated by the following example.

Example. A pump is to be selected similar to the pump of Fig. 4.9 to deliver 75 gal per min against a 65-ft head. It is desirable to operate it at a point on its performance curve comparable to the 150-gal-per-min point on the base curve, Fig. 4.9. The basic conditions are 150-gal-per-min capacity, 36-ft head, 7-in.-diameter runner, and 1760 rpm. They are q_1, H_1, D_1, and N_1, respectively. The required runner diameter from equation 4.24 is:

$$D_2 = 7(36^{1/4}/150^{1/2})(75^{1/2}/65^{1/4}) = 4.28 \text{ in.}$$

The required runner speed from equation 4.25 is:

$$N_2 = 1760(150^{1/2}/36^{3/4})(65^{3/4}/75^{1/2}) = 3870 \text{ rpm}$$

The power required can be calculated from equation 4.23, which, transposed, is

$$\text{hp}_2 = \text{hp}_1(N_2^3/N_1^3)(D_2^5/D_1^5)$$

or,

$$\text{hp}_2 = 1.63(3870^3/1760^3)(4.28^5/7^5) = 1.48$$

In equations 4.23, 4.24, and 4.25 the subscript-1 values *must* be taken from performance data for the basic pump and any subscript-1 value fixes all the other subscript-1 values. Subscript-2 values are not so fixed and may vary at random.

H in these equations is defined as total head in feet of fluid. Since these equations are based on geometric similarity and the terms are in ratio in each equation, H can be expressed in any convenient dimension, such as inches of mercury, inches of water, pounds per square inch, and can represent static head or pressure or velocity head as appropriately as total head.

4.11. Pump Performance on a System. The system to which a pump is attached could be made up of lengths of pipe, valves, various joints, orifices, transitions, etc. The sum of the resistances of the various elements, the elevation or fluid lift, and the velocity pressure head is the total head, which can be calculated by the Bernoulli equation (Sect. 2.6). A graph of

total head plotted against the capacity or rate of flow through the system is called a system characteristic curve, Fig. 4.14. The total head of this system is made up of 15 ft of elevation and pipe, elbow, and velocity fractions. The graph shows the rate of fluid flow which will result for various total pressures (heads) across the system.

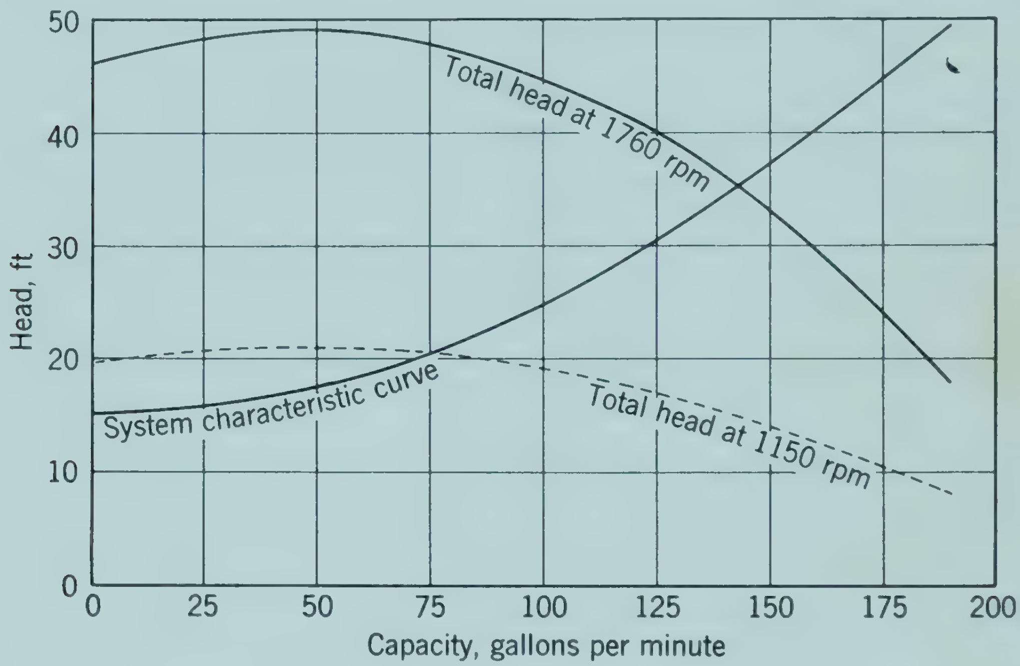


Fig. 4.14. Performance of a pump when attached to a specific system.

When a pump is attached to a system, the rate of fluid flow will depend upon the characteristics of the system and the characteristics of the pump. The point of operation can be determined by superimposing the system characteristic curve upon the pump performance plot. The intersection of the system characteristic curve with the total head curve defines the point of pump operation and the rate of flow through the system. Figure 4.14 shows such a plot for the pump of Fig. 4.9. Thus, the delivery rate is 140 gal per min; the power required, 1.8 hp; and the pump efficiency, 75 per cent.

4.12. Viscosity. Fluids that are pumped in processing work are frequently more viscous than water; milk, cream, oils, sugar solutions, molasses, for example. The relationship between viscosity and pump performance is not well defined, but certain important observations will help to solve pumping problems involving viscous fluids.

The efficiency of a pump decreases as the viscosity increases. The increased fluid friction between the pump parts and the passing fluid and between pump parts separated by fluid dissipates more mechanical energy as heat energy, and less of the shaft input energy is available to do useful work. It is not possible to provide a general correction procedure for the effect of viscosity upon efficiency since the loss in the pump is due to hydraulic friction and mechanical friction, which are not generally related.

The working head increases as the viscosity increases. Reynolds number, equation 2.6, varies inversely as the viscosity. Since a large Reynolds number is desired for most satisfactory performance, fluids of high viscosity must be moved in large-diameter pipes in order to minimize friction head losses. The rate of flow can be reduced to produce a further reduction in head loss.

REFERENCES

1. Abramson, E. L. Performance Characteristics of Turbine Pumps at 3500 and 1750 rpm. *Power* 91:300–301. May, 1947.
2. Addison, H. *A Treatise on Applied Hydraulics.* Third Edition. Wiley. 1945.
3. Church, A. H. *Centrifugal Pumps and Blowers.* Wiley. 1944.
4. Gosline, J. E., and O'Brien, M. P. The Water Jet Pump. University of California Publication in Engineering 3:167–190. 1934.
5. *Hydraulic Institute Test Code, Centrifugal Pumps,* Section F. Hydraulic Institute. 1936.
6. Perry, J. H. *Chemical Engineering Handbook.* McGraw-Hill. 1934.
7. *Test Code for Centrifugal and Rotary Pumps Power Test Codes.* A.S.M.E. 1927.

PROBLEMS

1. A whole milk homogenizer operating at a pressure of 2500 sq in. delivers 6 gal per min. If the pump efficiency is 82 per cent, what size motor is required? What size motor is required if the pressure is 1500 lb per sq in.? Pipe friction may be neglected. Estimate the velocity through the homogenizing valve.
2. A 1750-rpm centrifugal pump with a 4.75-in. impeller delivers 140 gal per min against a 20-ft water head and uses 1 horsepower. What is the pump efficiency? What is the head at complete shut-off?
3. The pump of problem 2 is to operate against a 25-ft head without changing efficiency. Specify the speed, discharge rate, and power required.

4. The pump of Fig. 4.9 is connected to 40 ft of 2-in. galvanized-iron pipe. The lift is 25 ft. The system contains 2 elbows, pumps from a tank, and discharges from the pipe. What is the water pumping rate?
5. Specify a pump geometrically similar to that of Fig. 4.9 to operate at maximum efficiency at a head of 40 ft and capacity of 180 gal per min. Impeller diameter, speed, and horsepower are required.
6. Determine the efficiency of the pump of Fig. 4.14 when operating at 1150 rpm.

CHAPTER 5

Fans

NOMENCLATURE

D = diameter, in.
H = total head, ft.
N = revolutions per minute.
P = fan pitch, ft per rev.
q = air rate, cu ft per min.
r = radius, ft.
α = fan-blade angle of twist, degrees.
γ = specific weight, lb per cu ft.

Fans are used in agricultural processing in connection with drying, ventilating, heating, cooling, refrigerating, aspirating, elevating, and conveying. Processing and other agricultural activities requiring fans is increasing. Costs are becoming more and more important, and it is necessary that the processing engineer be able to select and apply the best fan for any installation, taking the economic factors into consideration.

The terms fan, blower, compressor, etc., are frequently used interchangeably. The American Society of Mechanical Engineers [6, 7] has placed these devices into the following classifications:

Class I, *Compressors.* Operate at pressures equal to or *more* than *one* lb per sq in. (27.7 in. of water). Machines in this class are also called centrifugal compressors, turbocompressors, and blowers.

Class II, *Fans.* Operate at pressures of *less* than *one* lb per sq in. (27.7 in. of water). Machines in this class are also called centrifugal fans, fan blowers, or exhausters.

This classification was developed to expedite testing, the main difference between classes being the fact that the heat of compression and the variation in specific weight must be recognized and considered in Class I whereas in Class II it is of minor importance and may be neglected in most tests. Compressors are usually applied to an agricultural processing job as packaged units such

as air or refrigeration compressors. Consequently, a detailed treatment of compressors is not important in this book. Fans will be discussed since they must be selected, adapted, and perhaps designed for specific installations.

Fans may be classified as to type or design according to the following schedule, which is recognized by the National Association of Fan Manufacturers.[5]

5.1. Axial-Flow or Propeller Fans. In this type air flow is parallel to the shaft or axis. Propeller fan is a generic term. Technically axial and propeller fans are the same; past experience and general usage, however, have segregated the duties for which each fan is used.

5.2. Propeller Fans. This type may have two or more blades which may be of sheet steel or airfoil shape. The blades may be narrow or wide. They may have uniform or varied pitch. This type of fan, such as shown in Fig. 5.1, has been developed and used to handle large volumes of air against free delivery or low heads.

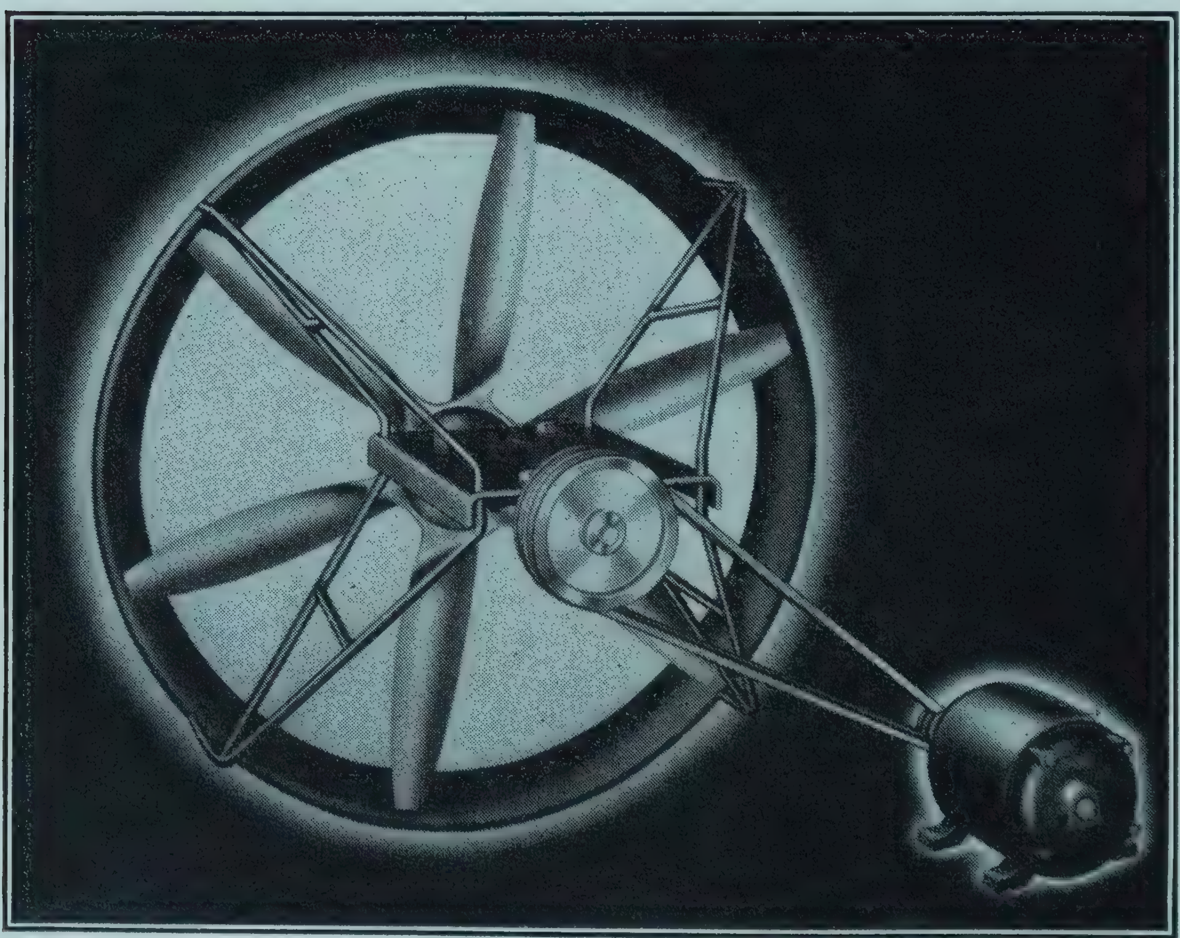


Fig. 5.1. A propeller fan. (*Courtesy* Hartzell Propeller Fan Co.)

One type of propeller fan is loosely described as a *disc fan* (Fig. 5.2). The disc fan has a blade area that covers an appreciable portion of the whole wheel area, and the center of the wheel, or hub, is of appreciable size. Because of these two details, the fan will operate against resistances slightly higher than the general line of propeller fans.

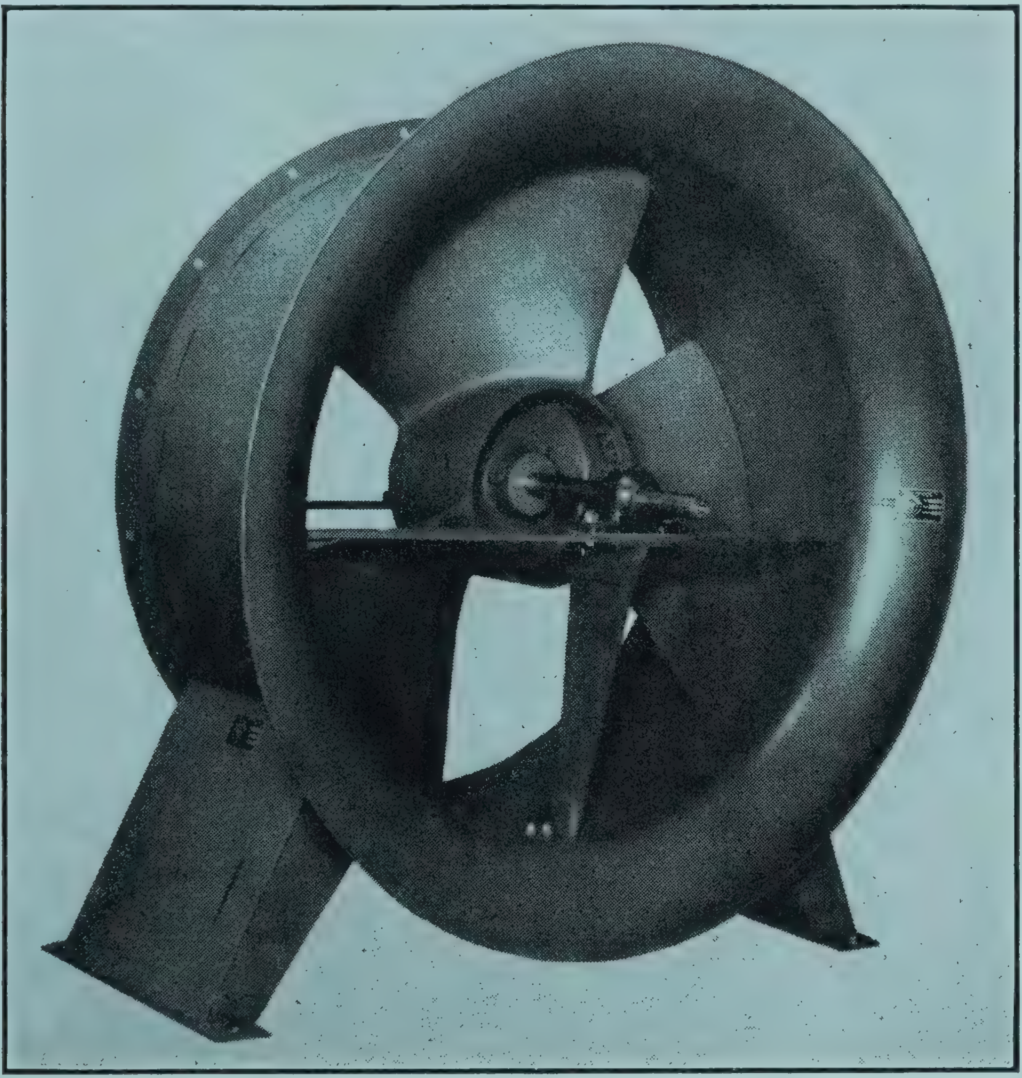


Fig. 5.2. A tube-axial fan, sometimes called disc fan because of the rotor shape. (*Courtesy* Westinghouse Corp.)

5.3. Axial-Flow Fans. These fans are similar to disc fans but are more refined. The hubs have been enlarged. The blade is warped for better efficiency, and the blades have a close radial clearance with the housing. As a result, they will operate against higher pressures and, because of the refinements, have a better efficiency.

Axial-flow fans are subdivided by the N.A.F.M. Code [5] thus:

5.4. Tube-Axial Fan. "A tube-axial fan consists of an axial-flow wheel within a cylinder and includes driving-mechanism supports either for belt drive or direct connection." (Fig. 5.2.)

5.5. Vane-Axial Fan. "A vane-axial fan consists of an axial-flow wheel within a cylinder, a set of guide vanes located either

Fig. 5.3. A vane-axial fan. Note the guide vanes behind the fan. (*Courtesy* The Buffalo Forge Co.)

before or after the wheel, and including driving-mechanism supports for either belt drive or direct connection." (Fig. 5.3.)

5.6. Centrifugal or Radial-Flow Fan. This type is shown in Fig. 5.4 and consists of a wheel or rotor within a scroll spiral type housing. The air enters parallel to the shaft, makes a 90° turn

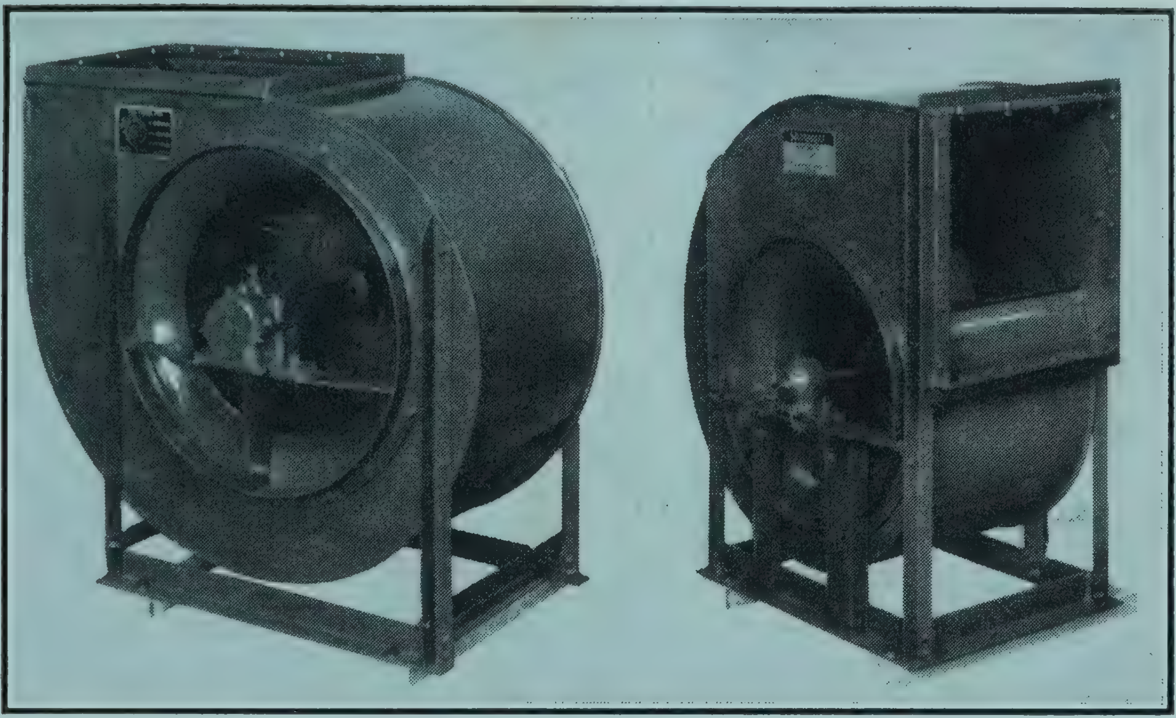


Fig. 5.4. Forward-curved-blade and backward-curved-blade centrifugal fans. (*Courtesy* Westinghouse Corp.)

in the fan wheel, and is discharged from the wheel (and housing) in a radial manner.

Centrifugal fans can be subdivided into the three classes shown in Fig. 5.5, which are discussed in the following sections.

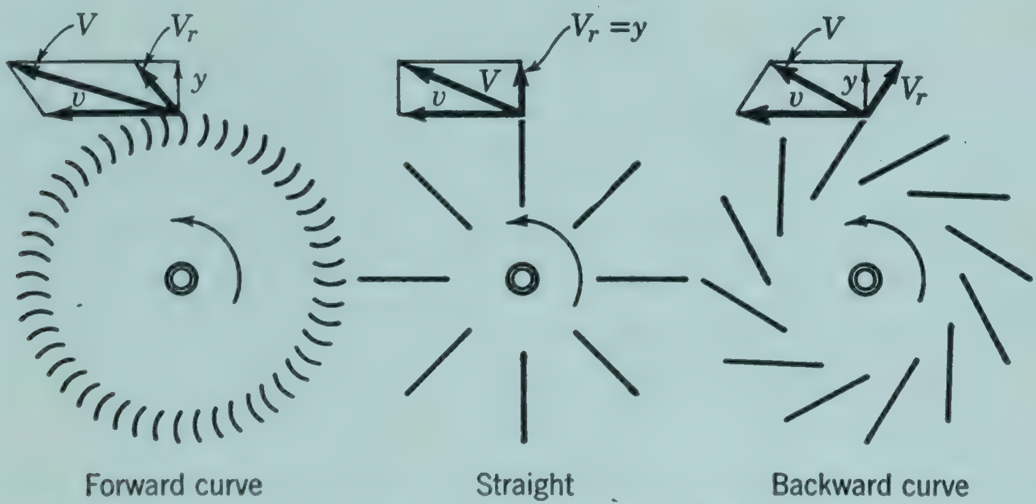


Fig. 5.5. The three types of centrifugal fan rotors, with velocity diagrams.

5.7. Type *a*, Forward-Curved-Blade Fans. This type has a rotor similar to a squirrel cage and a large number of blades, i.e., up to 60, narrow in the radial dimension but wide parallel to the shaft and facing forward in the direction of rotation like

a scoop. It is a low-speed fan, capable of operating at several inches pressure under most conditions but is limited to handling clean air.

5.8. Type *b*, Radial-Tip Straight or Double-Curved-Blade Fans. This type has a smaller number of blades—from 6 to 20—and the blades are essentially in a plane radiating from the shaft. The blades are normally about 2 to 3 times as long radially as they are wide. This type of fan usually has a larger housing than the other types and is more expensive; however, its price is justified by its ability to handle dirty air and to convey materials that go through the fan or to develop pressures beyond the range permissible with lighter weight fans.

5.9. Type *c*, Backward-Curved-Blade Fan. This type has about 12 blades, essentially flat and tilted backward from the direction of wheel rotation. It is inherently a high-speed type of fan with a self-limiting horsepower characteristic (Sect. 5.12). It is the most efficient of the various types of centrifugal fans and more expensive than the other types. Size for size, however, it has comparable efficiency and cost. With the added feature of the self-limiting horsepower characteristic, it is the best selection for reasonably clean air. It cannot as yet be recommended for dirty air.

FAN THEORY

The propeller fan is essentially an air screw. The twist or angularity of the blades is called "pitch," and theoretically it is the distance the air would be moved when turning the rotor 1 rev. If α is the angle of fan-blade twist, the "pitch" at any cross section at a distance r from the axis is

$$P = 2\pi r \tan \alpha \tag{5.1}$$

Note that the pitch increases as the radius for a fan with a constant blade twist or angularity. Consequently, the air near the tip of the fan is being moved at a faster rate than the air nearer the axis, the speed being theoretically proportional to the distance from the axis. Therefore, when the fan is operating against a material static head, air is forced back through the fan near the hub and recirculation or turbulence occurs as shown in Fig. 5.6 and lowers the efficiency.

Recirculation can be reduced or eliminated by warping the blades so that the pitch is constant. The warp design can be determined from equation 5.1 by holding P constant and determining α for variation in r. A large streamlined hub as shown in Fig. 5.3 improves performance by eliminating the region of recirculation. A cylindrical housing improves performance by guiding the stream of air that passes through the fan. The performance and efficiency of the axial-flow fan can be further improved by the addition of straightening vanes behind the rotor. The discharge from a "propeller" fan is in the form of a spiral, the rotation being in the same direction as the rotation of the fan wheel. Straightening vanes catch the air, turning it so its flow is parallel to the shaft. Elimination of turbulence reduces noise and improves the efficiency. Straightening vanes should be 2 to 4 in. from the blade wheel to minimize noise.

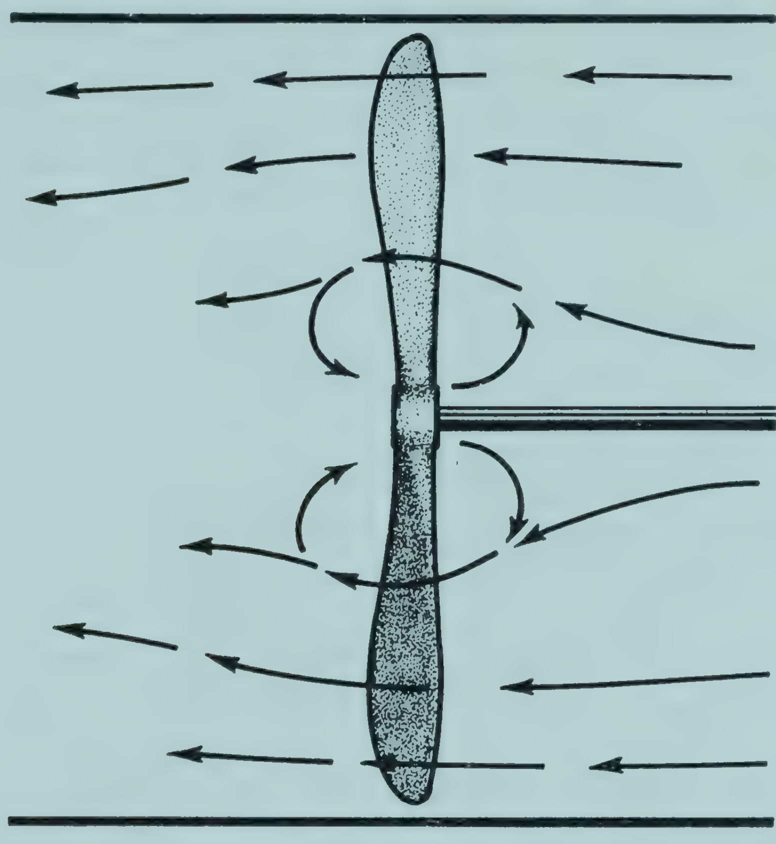


Fig. 5.6. Recirculation that results when propeller or disc fan is operated against too great a static head.

Pressures in excess of 60 in. have been developed by a single axial-flow fan, with a total efficiency of more than 85 per cent.

Centrifugal fan theory parallels that of centrifugal pumps (sect. 4.5 and Fig. 4.8) and will not be discussed here. However, con-

sider Fig. 5.6, which shows the three general types of centrifugal fan wheels in view of the centrifugal pump theory. Consider the velocity diagrams of the three general types shown in Fig. 5.5. For a constant peripheral speed, observe that the discharge velocity V for the forward-curved fan is approximately three times as great as for the backward-curved fan. Considering equation 4.13, note that the static pressure produced by the forward-curved fan is very small when compared with the backward-curved unit. This characteristic is outstanding as regards centrifugal fans and will be discussed in the following sections.

PERFORMANCE

5.10. Axial-Flow Fans. Fans are tested and rated on the basis of the Test Codes of the American Society of Mechanical

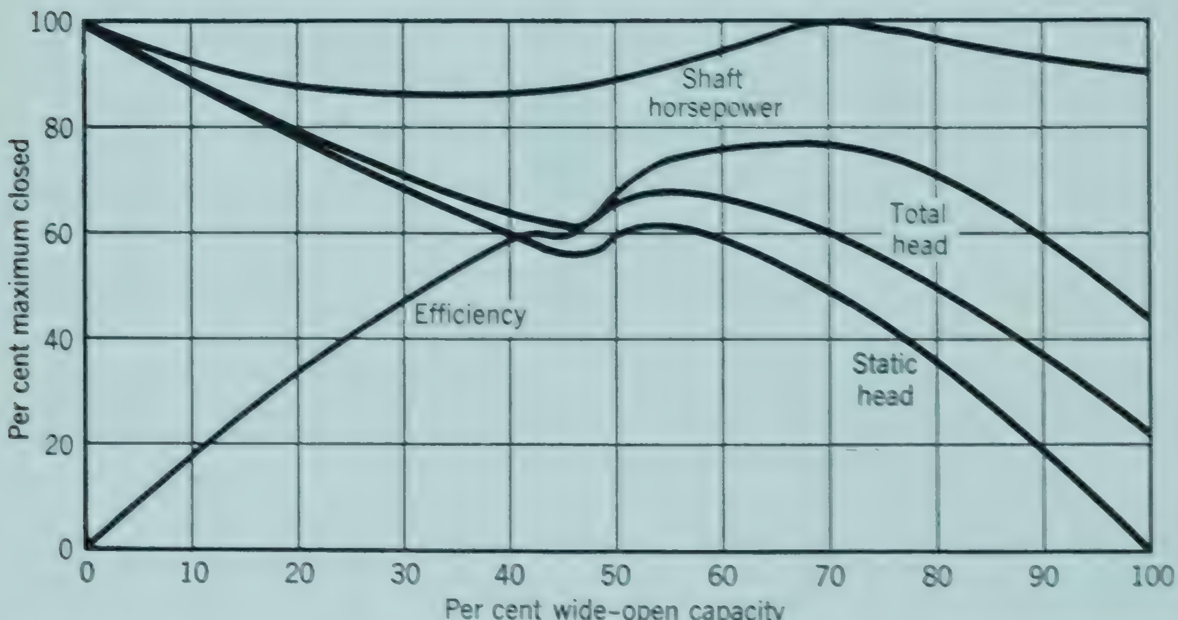


Fig. 5.7. Representative axial-flow-fan performance curves expressed on a percentage basis.

Engineers,[6,7] using the same procedure as discussed for centrifugal pumps in sect. 4.6.

A characteristic performance curve for an axial-flow type of propeller fan is given in Fig. 5.7. The reversal of the power and head curve is characteristic of propeller-type fans, although it is more pronounced in some particular makes. The point of reversal indicates the limit of stable operation of the fan. As a general rule, the power curve is relatively flat, especially within

the practical operating range, say above 40 per cent of wide-open capacity. When this rule does not hold, the power may decrease toward the wide-open capacity and increase near the point of complete shut-off. Under such conditions, the power requirement increases when the fan is throttled, owing to greater resistance head, and, unless extra power is available for this contingency, the power unit may be overloaded and difficulties may result.

It is generally advisable to assume that the power increases as the capacity decreases, although the rate of increase may be small and a reversal in the power curve usually exists.

5.11. Forward-Curved-Blade Fan. Fans of this type can be used for installations where the operating conditions are rela-

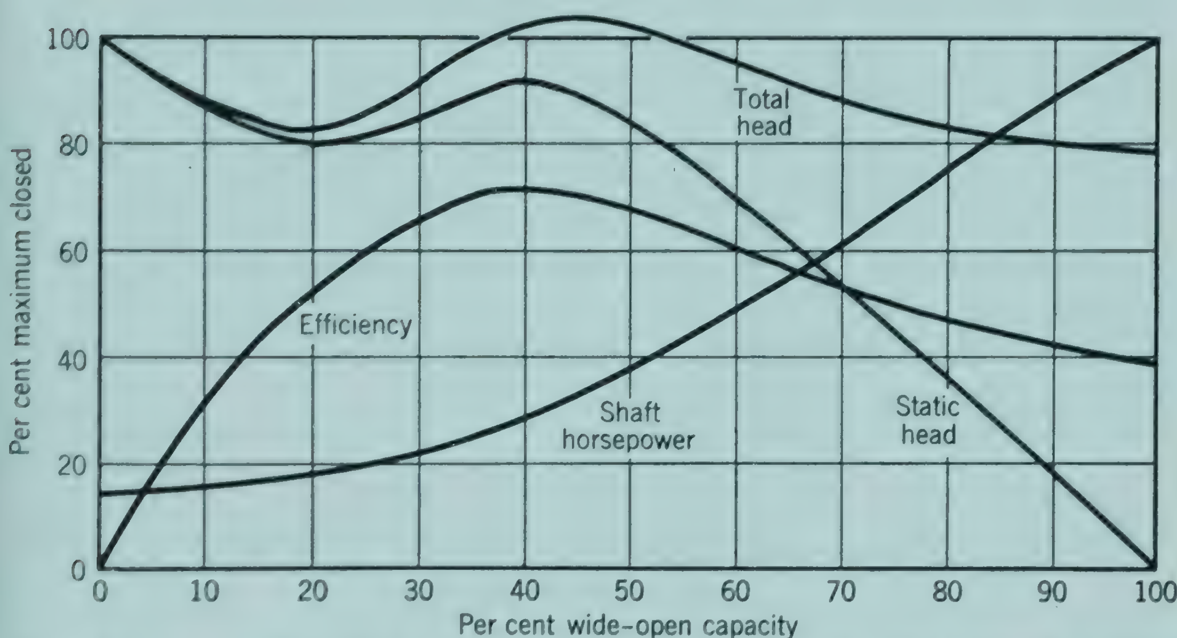


Fig. 5.8. Representative forward-curved-centrifugal-fan performance curves expressed on a percentage basis. The power curve is based upon the maximum rather than the closed value.

tively constant, static heads are low, and the air or gas is clean. Since the discharge velocity V is high, operating speeds can be low and the fan size small.

The performance characteristics of a forward-curved fan are shown in Fig. 5.8. Note that (1) the maximum efficiency occurs at 40 per cent wide-open capacity; (2) there is a complete reversal of head curves; and (3) the power requirement increases as the capacity increases.

Three capacities or rates of discharge are possible at the same static head. If the fan is connected to a system with a system

characteristic curve passing through this area, hunting may result between these three points.

The increase in power that develops as the capacity increases may be disadvantageous, since the motor might be overloaded when decreasing the system resistance.

In spite of the above-implied disadvantages, the forward-curved fan is suitable for ventilation and air conditioning jobs where the operating conditions are constant and the air is clean.

5.12. Backward-Curved Centrifugal Fans. The performance of this fan is comparable to the centrifugal pump since the design is similar.

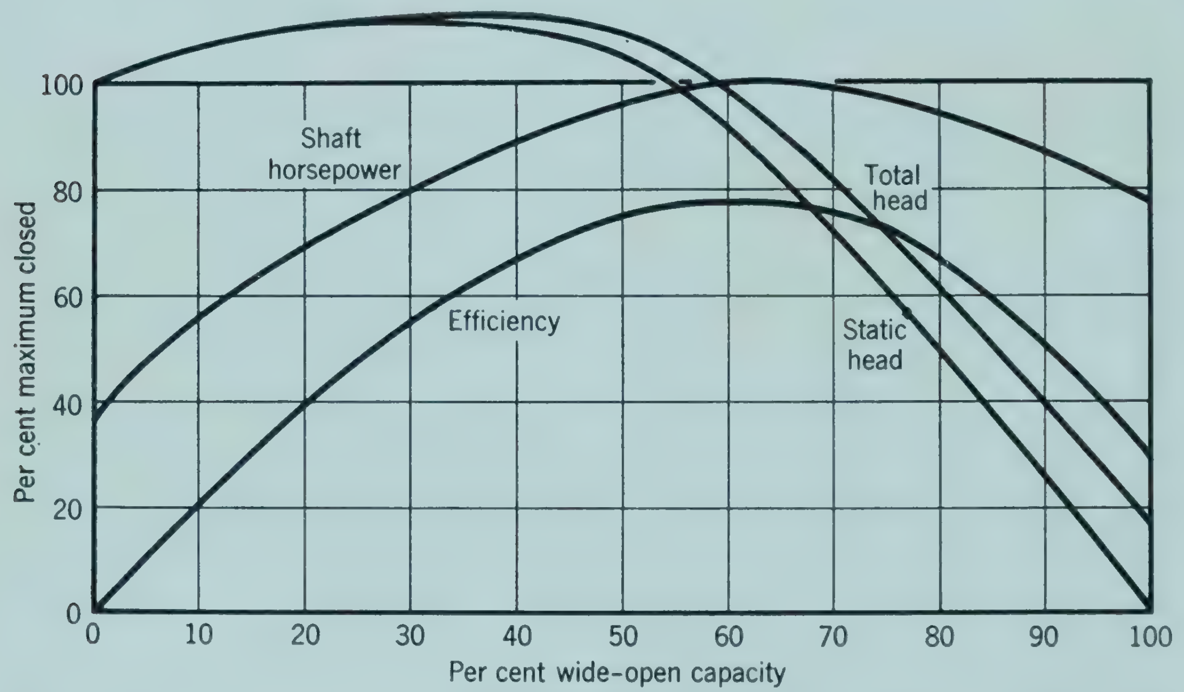


Fig. 5.9. Representative backward-curved-centrifugal-fan performance curves expressed on a percentage basis. The power curve is based on the maximum rather than the closed value.

Note from Fig. 5.9 that (1) the maximum efficiency occurs at about 60 per cent wide-open capacity; (2) the head curve increases fairly consistently from wide-open capacity to nearly complete shut-off; and (3) the power curve is a maximum at a point nearly coincident with the maximum-efficiency point.

The rising pressure curve practically eliminates the possibility of hunting. With the maximum in horsepower occurring at approximately the same point as the maximum efficiency, it is possible to pick the fan with peak efficiency and be unable to overload the motor by either increasing or decreasing the pressure or capacity as long as the speed is constant.

The backward-curved-blade fan operates at about 1.75 to 2.0 times the speed of the forward-curved-blade fan of comparable size and capacity.

5.13. Straight-Blade Fans. Their performance is similar to the forward-curved-blade fan in that they have a rising horsepower curve, although there is no complete reversal of the pressure curve. The efficiency is about the same. Air velocities through the wheel and in the housing are lower, as the fan is much larger. This fan was the first to be used in public buildings. When picked for peak efficiency, it is very quiet, but owing to its size it is also quite expensive.

Its use is now limited to pneumatic collecting or conveying where material must go through the fan or where the fan must handle dirty air or gas.

5.14. Combination-Curved-Blade Fans. This type covers fan blades of various shapes, some with forward-curved entering edges, some with backward-curved leaving edges, and some with combinations of radial sections.

Generally, these fans have performance curves similar to the forward-curved or straight-blade fans—that is, they have rising horsepower characteristics and peak efficiencies lower than those available with the backward-curved blade.

5.15. Factors Affecting Fan Selection. The following information must be known when selecting a fan. These factors aid in determining the type of fan to be selected and the size.

1. Quantity of air to be moved per unit of time.
2. Estimated system resistance and expected variations.
3. Amount of noise permitted.
4. Space available for fan.
5. Economic implications.

The quantity of air to be moved per unit of time will be determined from the type and size of installation and will not be discussed here.

The static pressure drop or resistance head can be determined conventionally by methods discussed in Chap. 2, Fluid Mechanics. Note that each type of loss, pipe friction, turns, valves, entrance, outlet, porous media, etc., varies nearly as the square of the velocity. Therefore, the total resistance or static pressure is

nearly a square function of the velocity through the system. Curves A, B, and C on Fig. 5.10 are such curves for three different systems and are called system characteristic curves. For example, system B requires a static pressure of 1.415 in. water to deliver

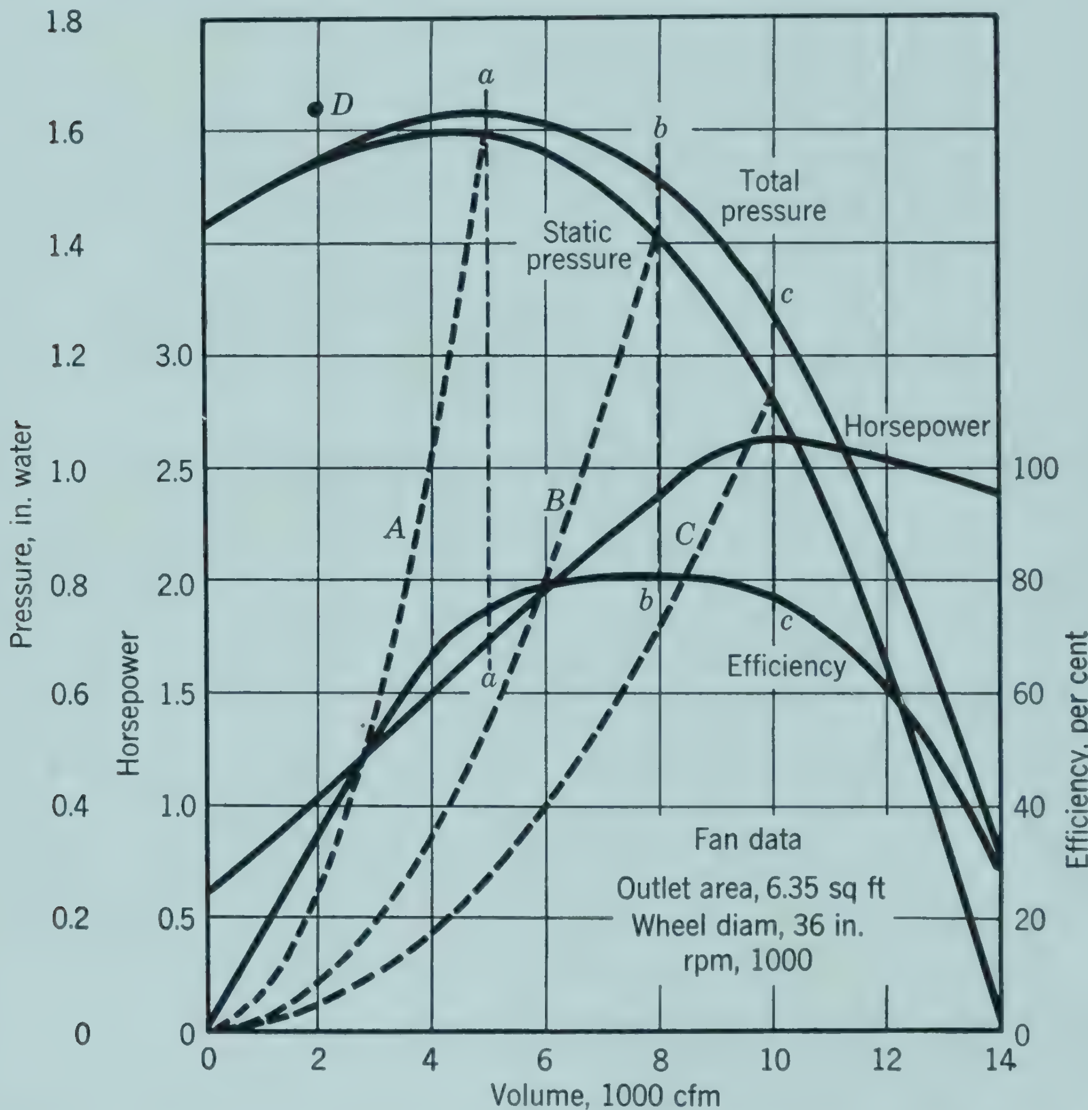


Fig. 5.10. Performance curves of a backward-curved centrifugal fan showing system characteristics.

8000 cu ft of air per min, but the static head would be only 0.8 in. water if 6000 cu ft per min were being delivered. The total pressure, power requirement, and efficiency are located on line b–b. The importance of these curves will be discussed in Sect. 5.16.

Noise is not as important a factor in processing work as in air conditioning or household ventilating, but since excess noise may indicate a significant loss of energy, noisy operation should be minimized as much as practicable.

There are usually a number of fans that will fulfill the requirements for a specific job. A large expensive fan operating at peak efficiency or a smaller, higher speed, less expensive, and less efficient fan will do a certain job, for example. Which to select will depend upon the amount of use, stability of use, and cost of power. Chap. 13 on cost analysis will aid in solving such problems as these.

5.16. Fan Selection. The system characteristic of a specified installation is curve *B* in Fig. 5.10. Eight thousand cubic feet of air per minute must be delivered. Consequently, the static operating head will be 1.41 in. of water. Significant variations in operating conditions are not expected.

If the factors causing resistance of a system do not change, any variation in the rate of air flow will be accompanied by a change in static pressure and the performance point will always be on the characteristic curve *B*. The operating point on a system characteristic curve will move only if the air-supplying device is varied or changed. A system of conduits for distributing conditioning air and a heat-exchanger fan are examples. If the system resistance is variable and the operating speed of the fan remains constant, the performance point will move along the static pressure curve. Suppose an installation with characteristic *B* contains an air filter that eventually becomes coated with dust that in turn increases the resistance of the system. The performance point would move along the static pressure curve from line *b–b* to *a–a* for example since the system characteristic would change from curve *B* to curve *C*. Note that the static head is greater, the volume flowing less, the efficiency less, and the power less. Suppose system *B* is a refrigerating system and contains a badly iced heat exchanger. When defrosted, the resistance will decrease and the performance point will shift along the static pressure line to *c–c*.

The rate of flow is frequently controlled by dampers in the system. Drying, air conditioning, ventilating, burner, draft, and aspirating operations are frequently so controlled. Dampers vary the system resistance and move the point of performance along the static pressure line. Although this control system is usually the most practical one, care must be exercised in design and application to avoid motor overloading on operations in the inefficient region.

A steep static curve is desired so that the variation in rate of flow will be small for a large change in static head.

If operating conditions are constant, a fan should be selected to operate at its point of maximum efficiency.

If operating conditions vary, selection should be made so that operation is in the most efficient range practicable. Generally, it is inadvisable to operate to the left of the maximum point on the static pressure curve because of possible unstable performance.

Note that the previous discussion is centered around static or resistance pressures and no mention is made of velocity pressure, which when added to static pressure gives total pressure. The velocity pressure indicates the energy needed to accelerate the air from zero to the velocity at the point of air passage through the fan. In most instances the velocity energy is lost at discharge so that the power required must be based upon the total pressure head.

Fan performance data are usually presented by the manufacturer in tabular form. The data represent only the practical operating range of the fan and include only such information as is needed to select a fan for a specific job.

5.17. General Performance. Fans perform in the same general manner as centrifugal pumps (sect. 4.6). Consequently, the following is true for any fan.

$$\text{Air horsepower} = q\gamma H/33{,}000 \tag{5.2}$$

where q = cu ft air delivered per min.
γ = specific wt, lb per cu ft.
H = total head, ft.

Total mechanical efficiency

$$= \text{Air horsepower}/\text{Shaft horsepower} \tag{5.3}$$

Some writers and companies use static efficiency, which is based upon static head, rather than total head. This practice may be justified since most fan jobs are defined in terms of static pressure and rate of air flowing. Furthermore, the velocity pressure is usually small as compared to the static pressure. However, since static efficiency is based upon static horsepower which is always less than total horsepower, the static efficiency is always less than

total or actual efficiency. Since the static efficiency is not a true index of maximum performance, its use is not recommended.

The pump laws, sect. 4.10, also apply to fans. Therefore, the equations that follow can aid in extrapolating performance data of geometrically similar fans. Although these laws are not perfectly applicable for widely varying conditions, the error is not significant.

The code of the National Association of Fan Manufacturers [5] permits published performance data to be secured by these mathematical procedures. The following excerpt from the code indicates the conditions under which this procedure is permissible.

"For larger size fans of the same design and similar proportions the performance may be calculated from tests obtained on fans having a wheel not less than 35 in. in diameter. For fans having wheels less than 35 in. in diameter the performance may be calculated from tests on fans of the same design and similar proportions and having wheel diameter not greater than the rated size."

$$D_2 = D_1(H_1^{1/4}/q_1^{1/2})(q_2^{1/2}/H_2^{1/4}) \tag{5.4}$$

$$N_2 = N_1(q_1^{1/2}/H_1^{3/4})(H_2^{3/4}/q_2^{1/2}) \tag{5.5}$$

$$\text{hp}_2 = \text{hp}_1(D_2^5/D_1^5)(N_2^3/N_1^3) \tag{5.6}$$

Since all the similar terms are in ratio, the dimensions need be consistent only for similar terms. Although H is defined as total head and has the dimension of feet of fluid being moved, because H operates in a ratio in equations 5.4, 5.5, and 5.6, it can be in pounds per square inch, inches of water, or any other convenient term. Furthermore, static or velocity pressures may be used since the ratio between either of these and total pressure is constant because of geometric similarity.

The term geometric similarity implies that a performance plot of a certain fan will apply to all other geometrically similar fans, axial or centrifugal, simply by changing the abscissa and ordinate values of volume, power, and pressure, efficiency remaining constant.

In these equations, the subscript-1 values are taken from the performance curve and any one value fixes all the others. For example, in Fig. 5.10, if H_1 is 1.28, then q_1 is 10,000, D_1 is 36, and N is 1000. The importance of these equations is demonstrated by

the following examples based upon the performance plot of Fig. 5.10.

Example 1. What speed, wheel diameter, and power would be required to operate the fan on system B at 7000 cu ft per min? The required speed, equation 4.25, is

$$N_2 = 1000(8000^{1/2}/1.41^{3/4})(1.09^{3/4}/7000^{1/2})$$
$$= 880 \text{ rpm}$$

The required diameter, equation 4.24, is

$$D_2 = 36(1.41^{1/4}/8000^{1/2})(7000^{1/2}/1.09^{1/4})$$
$$= 36 \text{ in.}$$

The required power, equation 4.23, is

$$\text{hp}_2 = 2.38(880^3/1000^3)$$
$$= 1.62 \text{ hp}$$

Note that there is no change in diameter. This shows that the rate of flow through a fixed system may be varied by varying the speed of the fan. The point of performance remains on the system curve, and the efficiency remains constant, in this case, at 81 per cent.

Example 2. Determine the diameter, speed, and power of a geometrically similar fan to operate at point D with an efficiency of 78 per cent (to right of maximum efficiency).

The basic (subscript-1) conditions at the 78 per cent efficiency point are: H, 1.34; hp, 2.6, and q, 9600. The new diameter is

$$D_2 = 36(1.34^{1/4}/9600^{1/2})(2000^{1/2}/1.64^{1/4})$$
$$= 15.7 \text{ in.}$$

The new speed is

$$N_2 = 1000(9600^{1/2}/1.34^{3/4})(1.64^{3/4}/2000^{1/2})$$
$$= 2540 \text{ rpm}$$

The new power requirement is

$$\text{hp}_2 = 2.6(15.7^5/36^5)(2540^3/1000^3)$$
$$= 0.68 \text{ hp}$$

Although this smaller high-speed fan will satisfy the new conditions, the excessive speed may contribute to short life and excessive noise. Practically, it would probably be advisable to use a point to the right of that selected as a base. This would yield a larger, slower fan with slightly higher power requirement. Consideration of a different type of fan might be advisable.

REFERENCES

1. Baumeister, Theodore, Jr. *Fans.* McGraw-Hill. 1935.
2. Church, A. H. *Centrifugal Pumps and Blowers.* Wiley. 1944.
3. *Fan Engineering.* Buffalo Forge Co. Fifth Edition. 1948.
4. Heating, Ventilating and Air Conditioning Guide. A.S.H.V.E. 1943.
5. Standards, Definitions, Terms, and Test Codes for Centrifugal, Axial and Propeller Fans. *National Association of Fan Manufacturers Bull.* 110. 1950.
6. *Test Code for Centrifugal Compressors and Exhausters; Power Test Codes.* A.S.M.E. 1935.
7. *Test Code for Fans; Power Test Codes.* A.S.M.E. 1946.

PROBLEMS

1. A pressure of 1.5 in. of water is required to move air up through a bin of grain at 20 cu ft per (min sq ft) of floor. If the floor has an area of 175 sq ft, what fan horsepower is required assuming 75 per cent efficiency?
2. The air system C of Fig. 5.10, which is carrying 10,000 cu ft per min, is altered so that the resistance is less and the fan delivers 11,000 cu ft of air per min. Determine the speed at which the fan must operate to deliver exactly 10,000 cu ft per min, and the static pressure and horsepower. What is the percentage reduction in power requirement?
3. A fan geometrically similar to that of Fig. 5.11 must operate at 2.0 in. pressure, 6000 cu ft per min, and an efficiency of 75 per cent. Specify the wheel diameter, speed, and power required.
4. Determine the discharge velocity of the fan of Fig. 5.11 from the outlet area and from the velocity pressure, both at 10,000 cu ft per min.

CHAPTER 6

Size Reduction

NOMENCLATURE

C = a constant.
D = average dimension, in.
E = energy, hp-hr.
F.M. = fineness modulus.
L = representative dimension.
m, n = exponents.

The general term "size reduction" includes cutting, crushing and grinding, and milling. The reduction in size is brought about by mechanical means without change in chemical properties of the material, and uniformity in size and shape of the individual grains or units of the end product is usually desired but seldom attained. Such processes as cutting fruit or vegetables for canning, shredding sweet potatoes for drying, chopping corn fodder, grinding limestone for fertilizer, grinding grain for livestock feed, and milling flour are size reduction. Other processes could be listed.

Milling is a trade term used relative to the reduction of grain into meal or flour. Milling as an over-all process includes size reduction, hulling, scarifying, polishing, sorting, mixing, and, in some instances, certain chemical reactions. The term "milling" is also used in connection with sorghum manufacture, extraction of the juice with rolls being the operation to which the term applies. Flax, hemp, and ramie processing to separate the fiber is generally referred to as milling, probably because the machine used is similar to that used for juice extraction from cane. Consequently, it can be seen that the terms commonly and generally included under size reduction are varied and not necessarily descriptive of the activities they represent.

SIZE CHARACTERISTICS

The performance of a machine for reducing the size of material is characterized by the capacity, the power required per unit of

material reduced, the size and shape of the product before and after reduction, and the range in size and shape of the resultant product.

Therefore, in order to study performance, a method or methods must be available for determining the size characteristics of any material.

The size and shape of the individual grains in any mass of material will depend upon the physical characteristics of the material, its previous history, and the method of reduction. Furthermore, it is extremely improbable that the shape of even a small percentage of the grains would approximate any simple geometric figure. In theoretical studies it is customary to represent an irregular particle by an equivalent sphere, cube, or other geometric figure, surface area or volume being used as the basis of comparison. Deviation from the performance of the idealized shape is recognized by the introduction of empirical factors. A factor is applied which indicates the degree of fit. The application of this concept to the performance of small particles under various conditions is both interesting and useful, but time and space will not permit it to be discussed here.*

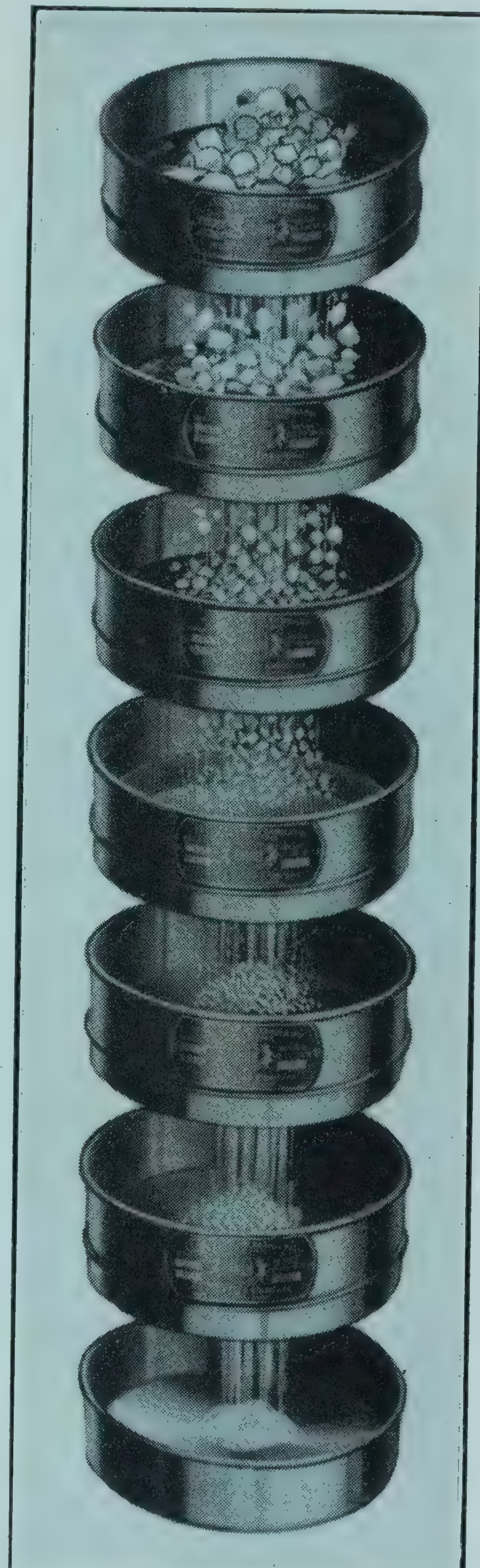


Fig. 6.1. Tyler sieves for classifying granular materials. (*Courtesy* W. S. Tyler Co.)

* For a thorough treatment of this subject, see Dalla Valle, J. M. *Micromeritics; The Technology of Fine Particles.* Pitman. Second Edition. 1948.

Reduced materials may be placed in three groups or classes based upon size.

1. Dimension range, particles or units which can be accurately measured and easily seen with minimum measurement approximately ⅛ in. or more. Diced fruit and vegetables and chopped forage are examples of this group.
2. Sieve range, particles with minimum dimension range of 0.125 to 0.0029 in. approximately. Granular materials such as ground feed and commercial fertilizers fall in this group.
3. Microscopic range, particles with minimum dimension less than 0.0029 in. Materials such as chemical powders, dusts, and Portland cement are examples.

This chapter deals with materials of the first two groups.

6.1. Tyler Sieves. The simplest method and the one most frequently used for placing granular materials in class 2 above is screening through a series of Tyler sieves, Fig. 6.1.* These sieves, which were originated in 1910, were adopted by the U. S. Bureau of Standards and are used as a basis for sizing all screened ma-

Table 6.1 TYLER STANDARD SCREEN SIEVES

Mesh, No. openings to inch	*Diameter of wire, in.*	*Size of Opening*	
		Actual	*Approx.*
....	0.148	1.050	1
....	0.135	0.742	$\frac{3}{4}$
....	0.105	0.525	$\frac{1}{2}$
....	0.092	0.371	$\frac{3}{8}$
3	0.070	0.263	$\frac{1}{4}$
4	0.065	0.185	$\frac{3}{16}$
6	0.036	0.131	$\frac{1}{8}$
8	0.032	0.093	$\frac{3}{32}$
10	0.035	0.065	$\frac{1}{16}$
14	0.025	0.046	$\frac{3}{64}$
20	0.0172	0.0328	$\frac{1}{32}$
28	0.0125	0.0232	....
35	0.0122	0.0164	$\frac{1}{64}$
48	0.0092	0.0116	....
65	0.0072	0.0082	....
100	0.0042	0.0058	....
150	0.0026	0.0041	....
200	0.0021	0.0029	....

* A U. S. Sieve series, similar to the Tyler sieves, was proposed by the National Bureau of Standards in 1919 and was adopted by the A.S.T.M.

terials used in processing. Their characteristics are given in Table 6.1.

The sieves listed in Table 6.1 constitute a normal set. The opening size is based upon the 200-mesh sieve, each opening being $\sqrt{2}$ or 1.414 times as large as the previous one. The openings are square, the size being the dimension of one side. Inter-

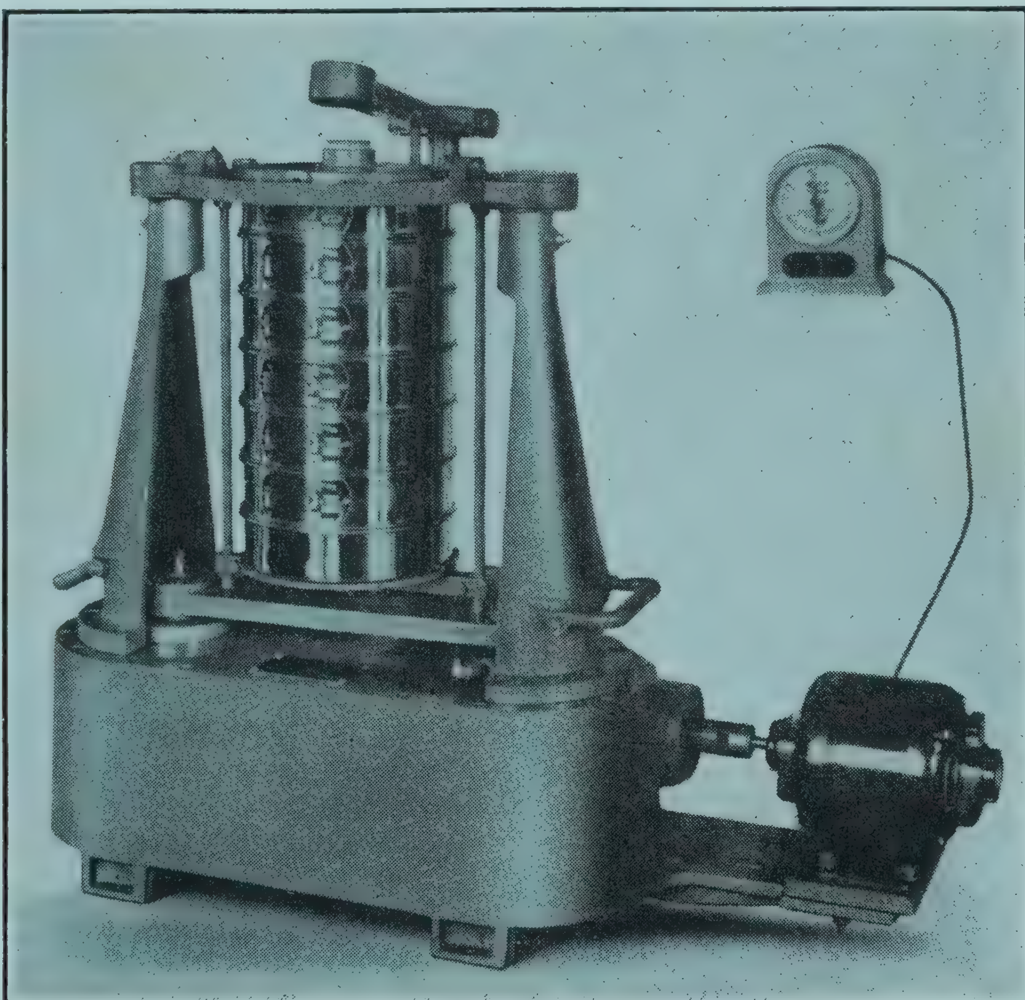


Fig. 6.2. A Ro-Tap machine used for a sieve analysis requiring precise results. (*Courtesy* W. S. Tyler Co.)

mediate sieves with opening ratios of $\sqrt[4]{2}$ or 1.189 are available and, if added, would constitute a complete set. Note that if every other screen is omitted in the normal set that each opening will be twice the previous one. This is an important feature as regards many size-reduction studies.

The techniques used in screening a sample have been standardized and should be followed if significant results are to be expected. The method and time of shaking are both important; recommended procedures should be consulted and followed when precise, significant data are required. A shaking machine, called a Ro-Tap, Fig. 6.2, which has a definite shaking motion and can

be adjusted for time of operation, can be used for carefully controlled studies.

The results of a screen analysis are reported in the accompanying tabulation in terms of percentage of material by weight remaining on each screen:

Mesh, in.	*Per Cent*
4	1
8	11
14	32
28	27
48	15
100	11
Pan	3

In this analysis, 32 per cent of the material by weight passed the 8-mesh sieve but would not pass the 14-mesh sieve. The size of the grains in this fraction varied from 0.093 to 0.046 in. in minimum dimension. This type of analysis can be used for classifying any granular material.

6.2. Fineness Modulus. A classification system devised by D. A. Abrams for concrete work is used by the American Society of Agricultural Engineers for determining the performance of feed grinders. The fineness modulus and uniformity index indicate the uniformity of grind or distribution of fines and coarses in the resultant product. The fineness modulus is defined as the sum of the weight fractions retained above each sieve divided by 100. The 3/8-in., 4-, 8-, 14-, 28-, 48-, and 100-mesh sieves are used in the set.

A simple method for determining the fineness modulus is shown in the tabulated example.

Tyler Screen			
Mesh	*Size of Opening*	*Per Cent of Material Retained*	*Multiplied by*
$\frac{3}{8}''$	0.371	1.0	7 = 7.0
4	0.185	2.5	6 = 15.0
8	0.093	7.0	5 = 35.0
14	0.046	24.0	4 = 96.0
28	0.0232	35.5	3 = 106.5
48	0.0116	22.5	2 = 45.0
100	0.0058	7.5	1 = 7.5
Pan		0.0	0 = 0.0
	Totals	100.0	312.0

$$\text{Fineness modulus} = \frac{312}{100} = 3.12$$

The standard procedure specifies a 250-g sample oven dried to constant weight at 212°F and shaken in the Ro-Tap for 5 min.

Note that if all the material were fine enough to pass through all the screens including No. 48 but would be retained on No. 100 the modulus would be 1.0. On the other hand, if all were retained on No. 4 screen, the modulus would be 6.0.

The average size of grain D in inches indicated by a modulus number F.M. can be calculated by the following equation,

$$D = 0.0041(2)^{\text{F.M.}} \tag{6.1}$$

which is shown graphically in Fig. 6.3.

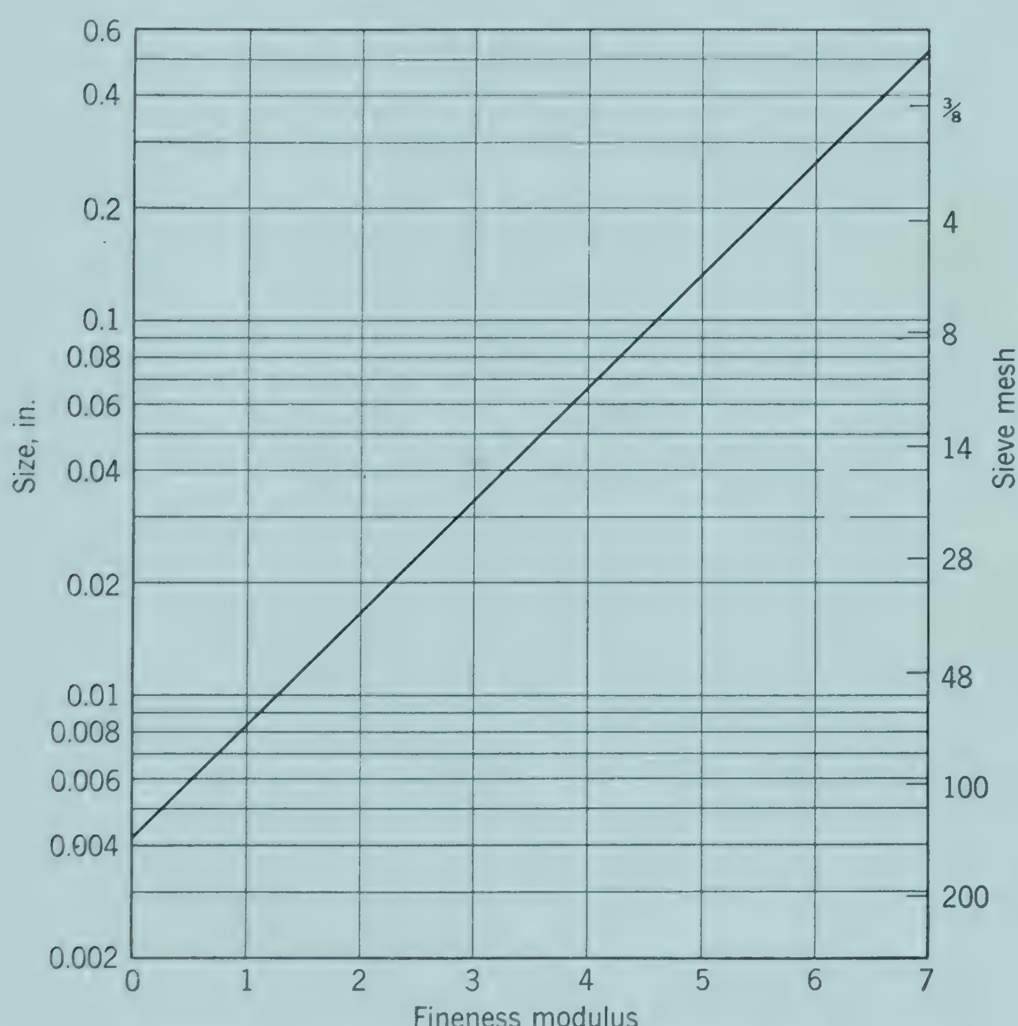


Fig. 6.3. Relationship between the fineness modulus and average particle size. Sieves are included for reference.

6.3. Uniformity Index. Although the fineness modulus gives an average size, it does not indicate the distribution of the fines and coarses in any sample, and the average grain size is not proportional to the modulus index. These objections can be overcome by using the uniformity index which is demonstrated on

the basis of the tabulated analysis. The ratio 1:6:3 which must total 10, determined as in the table, indicates the relative proportions of coarse, medium, and fine materials which are not indicated by the fineness modulus, 3.12. The recommended procedure is to quote both indices, thus: 3.12, 1:6:3.

A Screen (Mesh)		*B* Per Cent of Material Retained	*C* Totals Divided by 10	*D* Column C Rounded to Nearest Whole Number
$\frac{3}{8}$		1.0		
4		2.5		
8		7.0		Coarse
	Total	10.5	1.05	1
14		24.0		
28		35.5		Medium
	Total	59.5	5.95	6
48		22.5		
100		7.5		
Pan		0.0		Fine
	Total	30.0	3.00	3

Note that only two screens, numbers 8 and 28, would be required to secure a uniformity index if the fineness modulus were not required.

Value of Ground Feed. A great number of feeding tests of various feeds have been made with the several types of livestock and poultry to determine the value of grinding feed. These tests and accepted farm practice show that coarse grinding is advisable in most cases, specific over-all recommendations being as follows.

It is not advisable under any condition to feed finely ground grain to farm animals except small chicks, and for them the ground grain granules should not be powdery. Tests show that the fine material does not perform any better and in some cases not as well as larger granules. It is believed that the finely ground materials pass through the digestive tract too fast to be acted upon by the various digestive processes.

Grinding of forage is questionable as regards its feeding value. Coarsely chopped hay and stover may be advisable to minimize the amount of material thrown out of the feed bunks by the

animals, but there is little indication that the feeding value is increased. Chopping may be justified in order to mix a less palatable but nutritious forage with other feed. Fine chopping or grinding does not improve the forage and will probably lower its quality by exposure to oxidization. Also, the ability of the animal to digest the material will be decreased. Alfalfa and other forage crops for poultry and pigs must be ground fine to provide consistency for eating.

The classification of ground grains and forages into coarse, medium, and fine grades on the basis of modulus indices by Silver[15] is given in Table 6.2.

Table 6.2 FINENESS MODULUS FOR CLASSIFYING GROUND FEEDS

		Grind			
Material	*Whole Grain*	*Coarse*	*Medium*	*Fine*	*Very Fine*
Ear corn		4.80	3.60	2.40	1.80
Shelled corn	6.00	4.80	3.60	2.40	1.80
Barley	5.00	4.10	3.20	2.30	1.50
Oats	4.50	3.70	2.90	2.10	1.40
Soybeans	6.00	4.80	3.60	2.40	1.80
Wheat	5.00	4.10	3.20	2.30	1.50
Corn fodder	....	5.50	4.20	2.90	....
Hay	....	4.00	3.10	2.20	1.40

6.4. Energy Requirements. Consider the symmetrical particle in Fig. 6.4 which is to be reduced to symmetrical particles of a smaller size. Figure 6.4 could be represented as a cube, a parallelepiped, a sphere, or some other shape. The particles resulting from the reduction could be any shape or a great number of shapes.

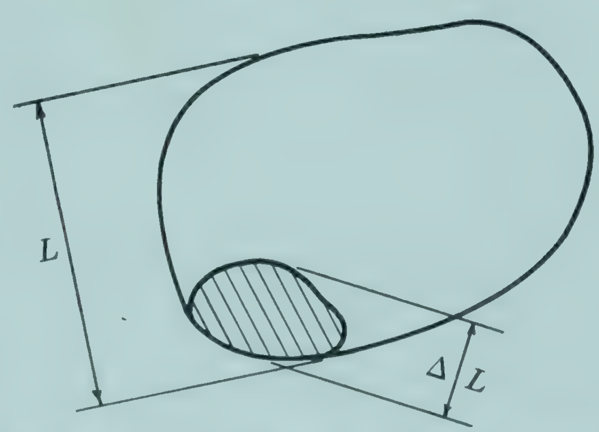


Fig. 6.4. An idealized particle which is to be reduced to geometrically similar particles.

The required energy must be related to some function of the initial and reduced particle, and, since the particles are assumed symmetrical, a common dimension would probably be used so that:

$$\Delta E \propto \Delta L^l/L^m \tag{6.2}$$

or

$$\Delta E = C\,\Delta L/L^n \tag{6.3}$$

This generalization implies that the energy required to reduce a unit is proportional to a dimension of the reduced particle relative to a similar dimension of the original particle raised to some power n.

Therefore, the energy necessary to reduce a specific mass of particles from one size to another is:

$$E = -C\int_{L}^{\Delta L} \frac{dL}{L^n} \tag{6.4}$$

Kick assumed that the energy requirements are a function of a common dimension of the material; so $n = 1$ in equation 6.3. Therefore, the energy requirements are:

$$E = C \ln (L_1/L_2) \tag{6.5}$$

Equation 6.5 is known as Kick's law.

Rittinger assumed that size reduction is essentially a shearing procedure. Consequently, the energy required is proportional to the new surfaces created, which in turn are proportional to the square of a common linear dimension. Therefore n in equation 6.4 equals 2 and the energy requirements are:

$$E = C\left(\frac{1}{L_2} - \frac{1}{L_1}\right) \tag{6.6}$$

This equation is known as Rittinger's law.

Integrating equation 6.4 gives a generalized relationship,

$$E = \frac{C}{1-n}\,({L_2}^{1-n} - {L_1}^{1-n}) \tag{6.7}$$

For example, if 5 hp-hr are required to reduce a material from ¼-in. size to 10 mesh, how much power would be required if the the reduction were to 20 mesh?

By Kick's law:

$$E = C \ln (L_1/L_2)$$

$$C = \frac{E}{\ln (L_1/L_2)} = \frac{5}{\ln (0.25/0.065)} = 8.56$$

$$\therefore E = 8.56 \ln (0.25/0.0328) = 7.55 \text{ hp-hr}$$

By Rittinger's law:

$$E = C\left(\frac{1}{L_2} - \frac{1}{L_1}\right)$$

$$C = \frac{E}{(1/L_2) - (1/L_1)} = \frac{5}{(1/0.065) - (1/0.25)} = 0.438$$

$$\therefore E = 0.438[(1/0.0328) - (1/0.25)] = 11.6 \text{ hp-hr}$$

Kick's and Rittinger's laws were developed from studies of materials common to the chemical and mechanical engineering areas, talc, coal, limestone, etc., being examples. Such materials are different from agricultural materials such as forage, small grains, and fertilizer components. Considerable deviation from these laws may be expected when studying reduction of such agricultural materials.

SIZE-REDUCTION PROCEDURES

The size of agricultural products is reduced by (1) cutting, (2) crushing, and (3) shearing either singly or in combination.

6.5. Cutting is separation or reduction which is produced by pushing or forcing a thin, sharp knife through the material to be reduced. Minimum deformation and rupture of the reduced particles results. The new surfaces that are produced by the sharp edge of the knife are relatively undamaged. Cutting is especially well adapted for reduction of fruit and vegetables. Since the pores in the new surfaces are open because of minimum damage from the sharp edge, drying or leaching or any process requiring transfer of a liquid or vapor to or from the material proceeds at a maximum rate.

The most satisfactory cutting device is a knife of extreme sharpness and as thin as structurally possible. The motion of the

knife should be such that the edge has a sawing component in moving through the material. This provides a smoother cut, probably with less energy.

6.6. Crushing is reduction by applying a force to the unit to be reduced in excess of its strength. Failure results by rupture of the material in many directions. The resulting particles are irregular in shape and size. The characteristics of the new surfaces and particles are dependent upon the type of material and the method of force application.

Limestone and other chemical fertilizers, ground feed for livestock, flour and meal, and fruit and vegetable purées are produced in part or whole by crushing. Crushing is used to extract juice from sugar cane and to break the structure of forage crops to speed drying.

The force used in crushing can be applied statically as is done when cracking a walnut in a vise or dynamically as with a hammer. Crushing by means of a rigid roll or bed such as the sorghum mill is an example of static force application. The hammer mill exemplifies dynamic force application.

6.7. Shearing is a combination of cutting and crushing. If the shearing edge is thin and sharp, performance approaches that of cutting. A thick, dull shearing edge performs more as a crusher.

Shearing is usually used for reducing materials of a tough fibrous nature where some crushing may be advantageous and the resulting units are of large uniform size. Cutting ensilage is an example.

The shearing units consist of a sharp knife and a bar. The knife is usually thick to withstand the shock that results when it hits the material. For best performance, the clearance between the bar and the knife should be as small as possible and the knife as sharp and thin as practicable.

REDUCING DEVICES

6.8*a*. Hammer Mills. The hammer mill is used for a variety of size-reduction or "grinding" jobs. Besides feed preparation, it is used for pulverizing limestone and the ingredients for commercial fertilizers. It also has many industrial applications.

A hammer mill consists essentially of a rotating beater and a heavy perforated screen, Fig. 6.5. The material is introduced into the housing, and the beater, which consists of a series of hammers turning at 1500 to 4000 rpm, beats and pounds the

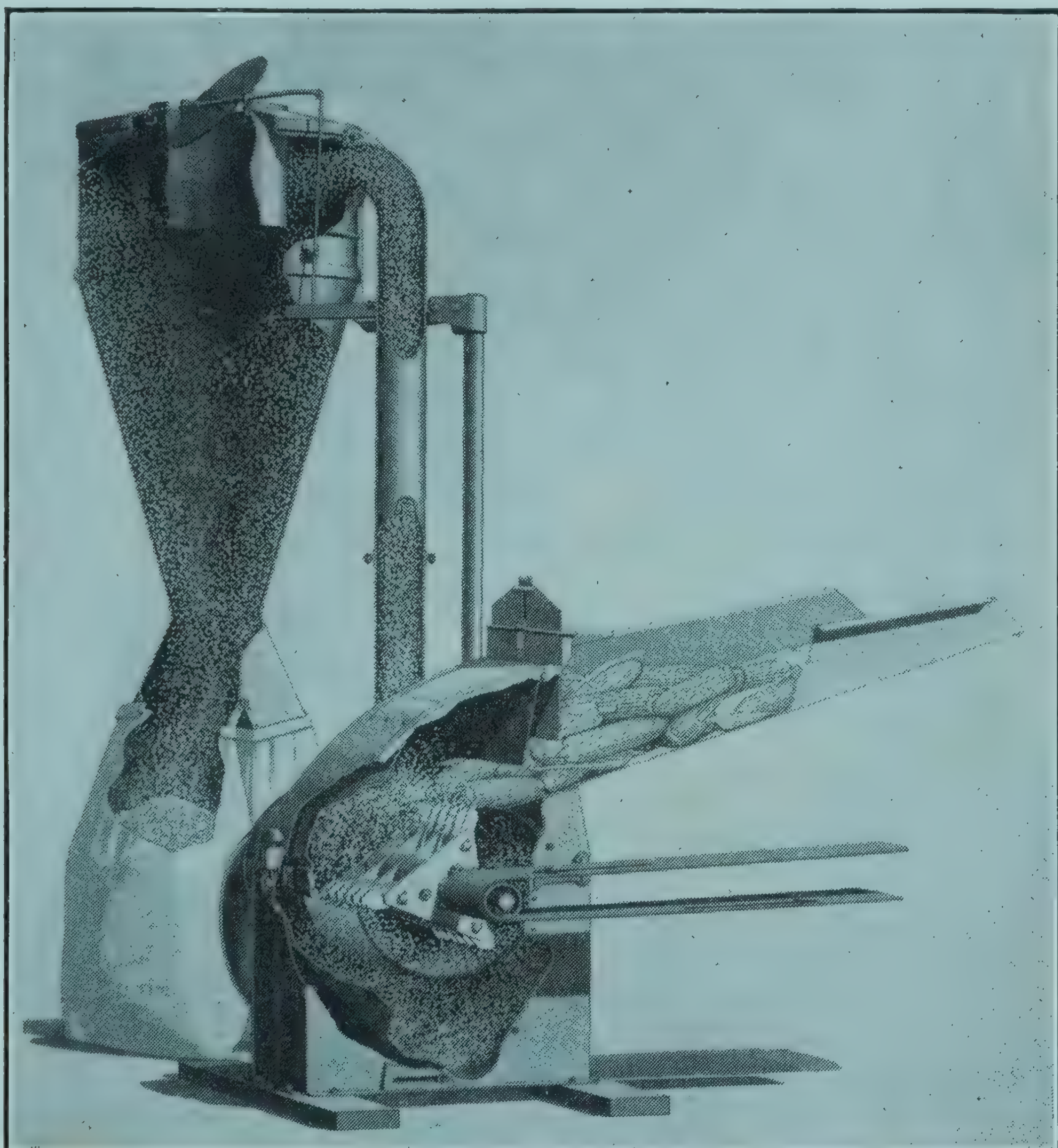


Fig. 6.5. Phantom view of a hammer mill with a cyclone for collecting the reduced material. (*Courtesy* John Deere Co.)

material until it is small enough to pass through the screen at the bottom. Fineness of division is controlled mainly by the size of holes in the screen, although the rotor revolutions per minute and the rate of feed are additional control factors.

The hammers are rigidly fixed to the shaft or swing as shown in Fig. 6.5. There is less danger of the swinging hammer causing damage if a large metallic object gets into the mill by accident. The striking edge of the hammer is designed in a great many ways, thus indicating that there is no one best design. The

swinging hammers are usually reversible so that two or perhaps four edges are available for use per hammer.

The hammer mill is assumed to reduce size by impact. The terrific speed of the hammer produces kinetic energy that is dissipated on the material, causing it to disintegrate. Although most of the size reduction is probably by impact between the material and the hammers, no doubt some shear between the screen or other parts of the mill and the particles takes place. The material is beaten and hammered until it is small enough to pass through the screen. After passing through the screen, it is removed by shovel, auger, chain elevator, or by a fan.

The advantages of the hammer mill are:

1. Simplicity.
2. Versatility.
3. Freedom from significant damage due to foreign objects.
4. Freedom from damage when operating empty.
5. Hammer wear does not materially reduce its efficiency.

Disadvantages *may* be:

1. Inability to produce a uniform grind.
2. High power requirements.

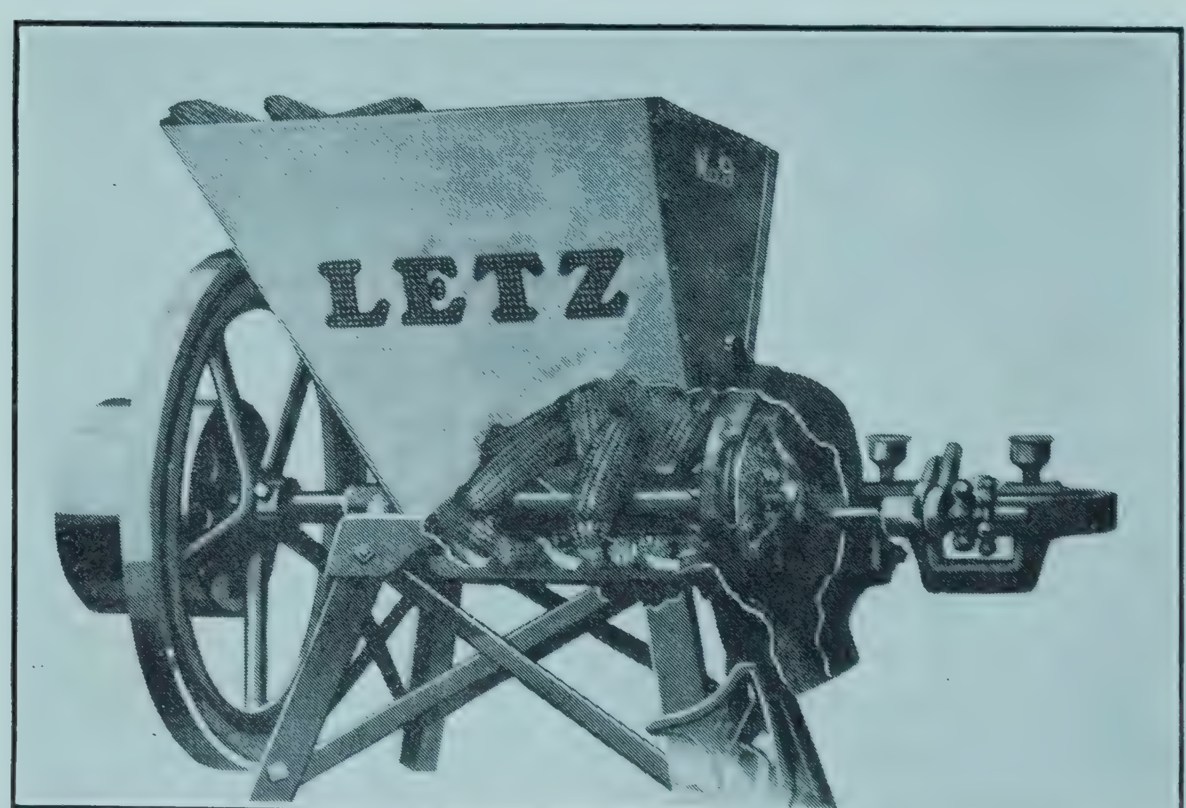
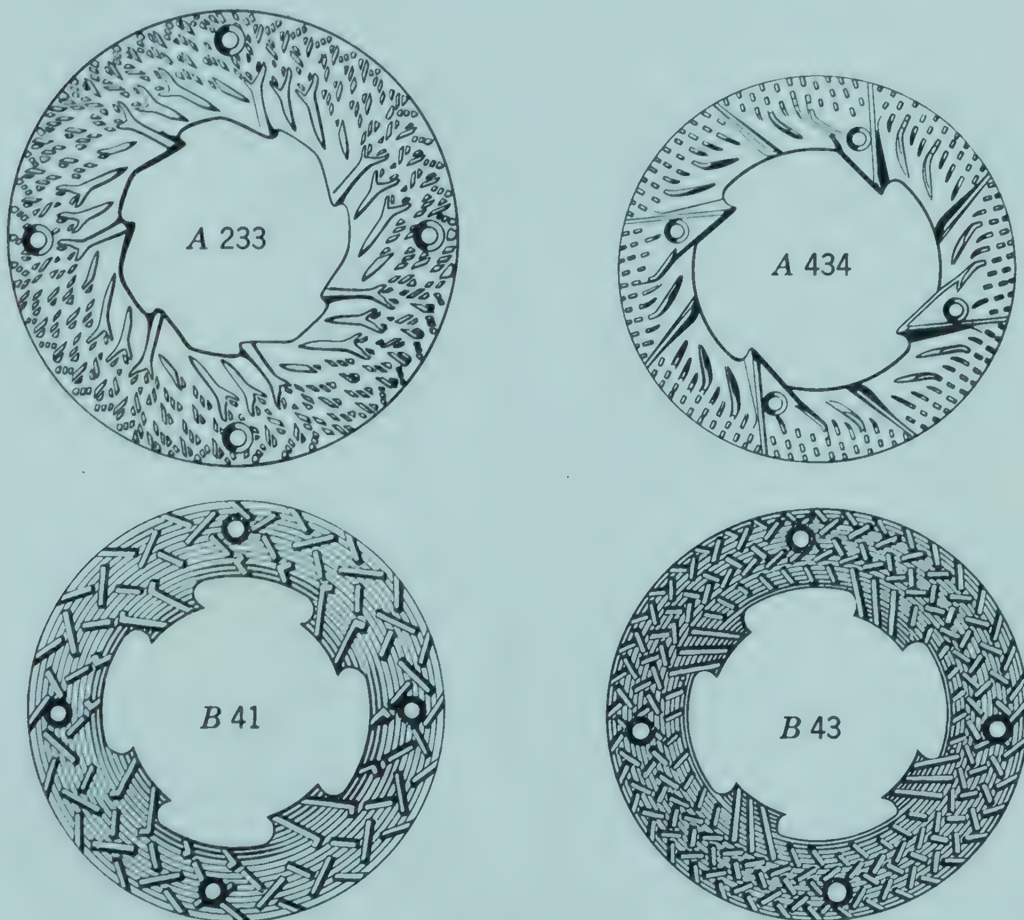


Fig. 6.6. Phantom view of a burr mill. (*Courtesy* Letz Manufacturing Co.)

6.8*b*. Burr Mills. Burr mills, also called plate mills, consist essentially of two roughened plates, one stationary, the other rotating, Fig. 6.6. The material is fed between the plates and is reduced by crushing and shear. If the material is fed slowly so the flutes are not filled, reduction is probably mainly by shear. With faster feed and flutes filled, both shear and crushing no doubt exist. Overfeeding reduces the effectiveness of the grinder, and excessive heating results. The plates are designed for a variety of jobs (Fig. 6.7) and are usually made of chilled cast iron although alloy steel may be advisable in certain cases. Operating speeds are usually less than 1200 rpm.



No. A-233: Medium fine for small grains, ear corn.
No. A-434: Medium fine for small grains, ear corn, roughage, high capacity.
No. B-41: Uniform coarse for small grains, ear corn.
No. B-43: Extreme fine for dry small grains, dry ear corn.

Fig. 6.7. Examples of "burrs" or "plates" for burr mills, a few of many designs for various duties. (*Courtesy* Letz Manufacturing Co.)

The fineness of reduction is controlled by the type of plates and by the spacing. The spacing screw is spring loaded so that the space will increase in case of an overload or if a foreign object

gets into the mill. Small rocks and metallic objects may not cause damage, but breakage can be expected if large objects are fed into the mill.

The attrition mill is a heavy-duty precision "burr" mill used in the commercial preparation of feed and food products. Each burr rotates and is driven independently, speeds are much higher, and design and construction are more precise.

The advantages of the burr mill are:

1. Low initial cost.
2. Product may be relatively uniform.
3. Power requirements may be low.

Disadvantages are:

1. Foreign objects may cause breakage.
2. Operating empty may cause excessive burr wear.
3. Worn burrs yield poor results.

6.9. Crushers. Crushers reduce the material by pressing or squeezing it until the material breaks. Crushing is an important industrial operation, and a variety of types of machines are in

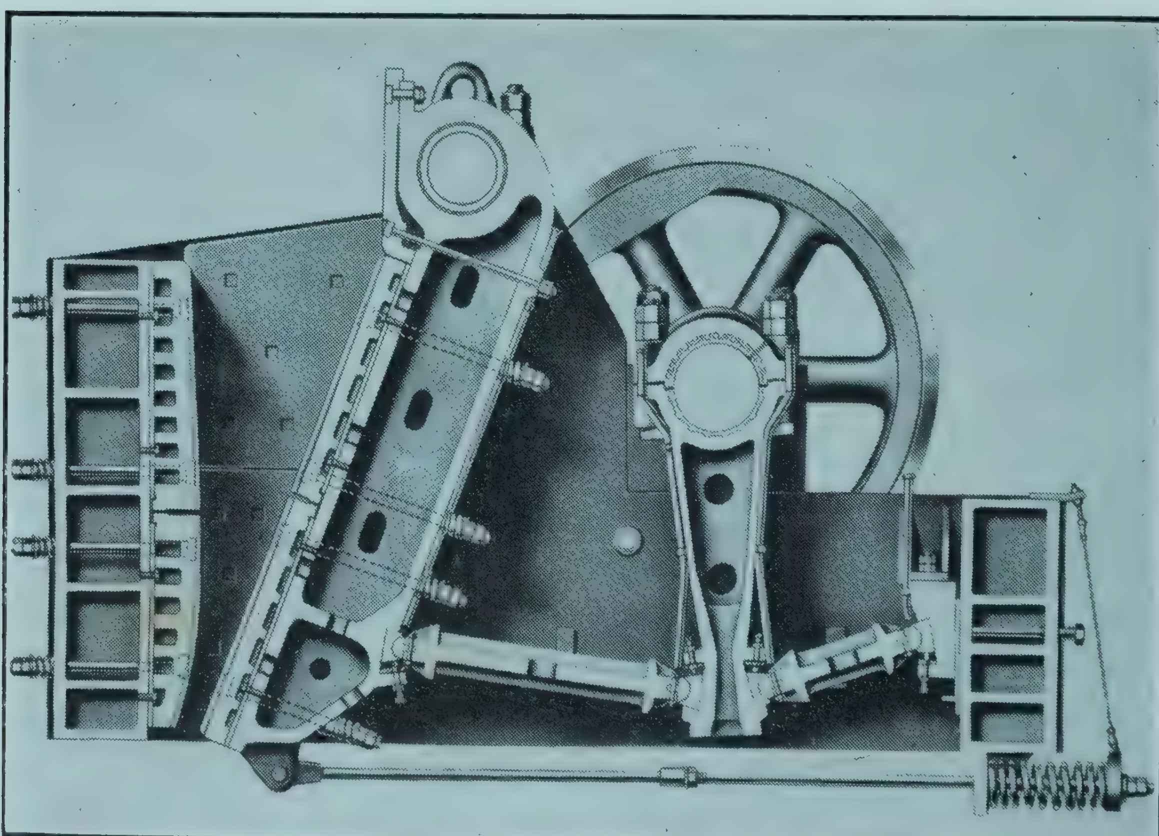


Fig. 6.8. Jaw and gyratory crushers. (*Courtesy* Allis-Chalmers Manufacturing Co.)

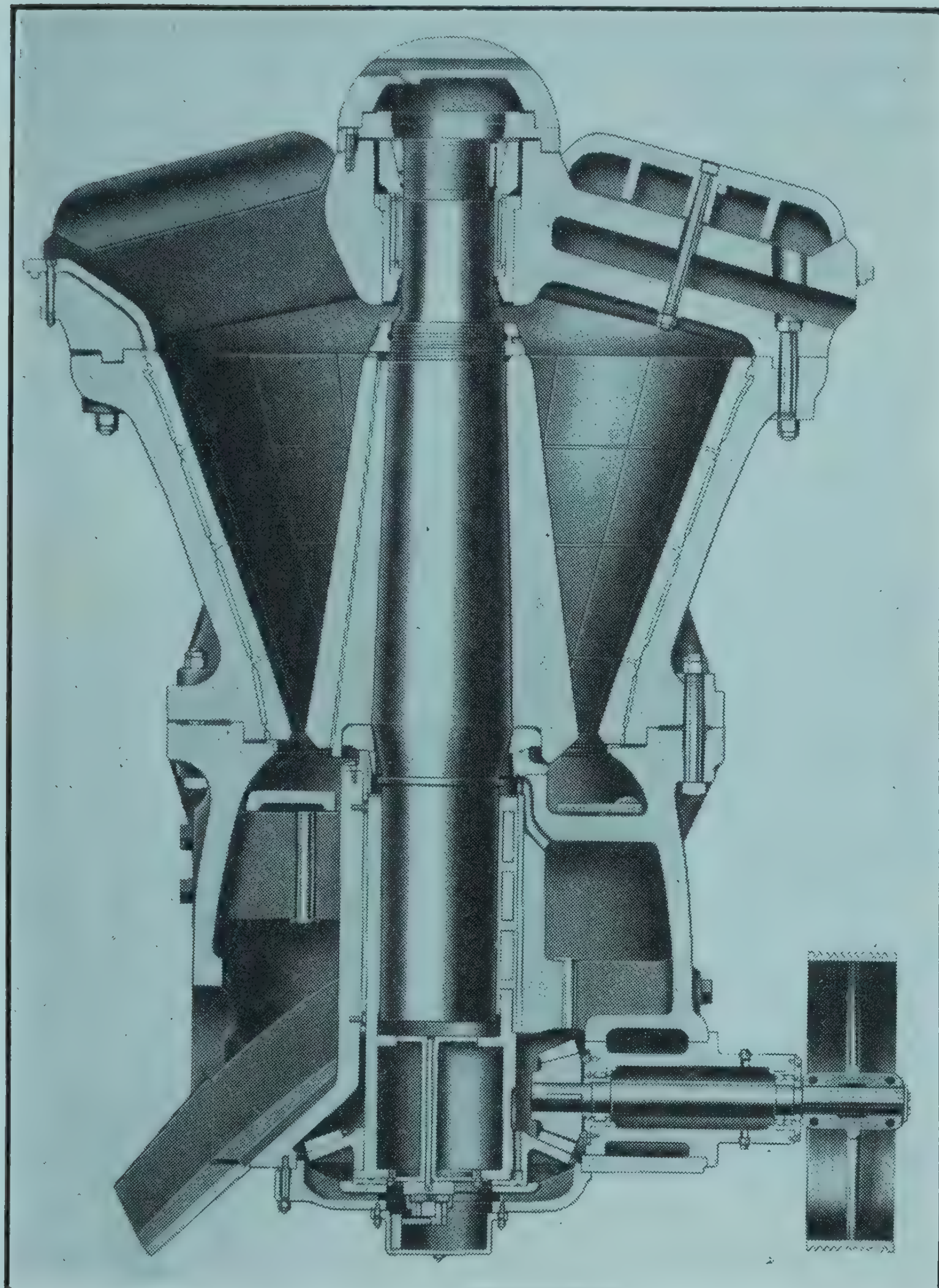


Fig. 6.8. (*continued*).

use. Agricultural application of crushers is important but not extensive. The important adaptations will be discussed briefly.

Lime and other stones are given an initial reduction by a jaw or gyratory crusher, Fig. 6.8. The jaw crusher is the cheaper and the slower of the two and is used for smaller operations. The shaft carrying the crushing cone in the gyratory crusher is free turning. It is loosely fitted at the top and is given a circulatory motion of small amplitude at the bottom.

The resultant motion of the cone crushes the material in a manner similar to the jaw crusher, but operation is smoother and the relative capacity is higher.

For primary reduction, roll crushers in various forms are used by themselves or, more frequently, in connection with burr or hammer mills. Burr mills are frequently combined with a single roller crusher so that large materials such as ear corn will be crushed to a suitable size for feeding between the plates. Grinders of this type are moderately versatile and are found in use on

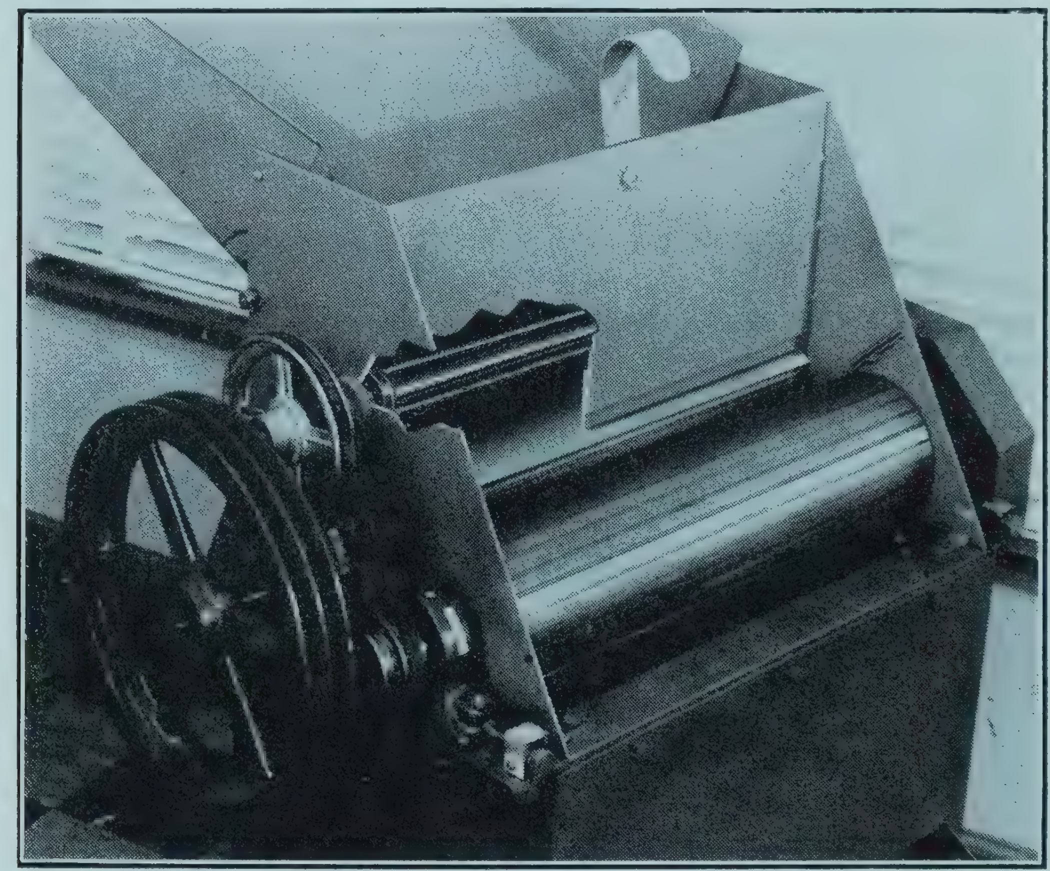


Fig. 6.9. Phantom view of a roller crusher for small grains. (*Courtesy* The Farnam Co.)

many farms. Double rollers, with or without serrated surfaces, produce a more uniform product than most other reducers. They are used extensively in the industrial preparation of cereals for human consumption. Figure 6.9 is a roller crusher for farm preparation of animal feed. The roller crusher for "grinding" grain on the farm was used to some extent years ago. Kable [9] in 1927 made this statement:

"Roller mills are passing out of use for farm grinding. The reason is largely the demand for a finer product than can be produced between rolls. The cost of roller mills is higher than burr mills but lower than hammer mills. The mills are durable and require little attention in operation. The product from the rolls

in grinding barley, wheat, and corn is not dissimilar to that from the burr mill except that the percentage of fine material obtainable is lower. Based upon the few tests made, the power required for rolling is somewhat greater than that used for the coarser grinding with burr mills. The continued use of roller mills will depend largely upon whether the coarser grinding will meet farm requirements."

It is interesting to note that the major objection to roller mills in 1927 is now a major asset in that the desired product should have few fines and many coarse particles. The possibilities of farm use of the roller crusher should be resurveyed.

PERFORMANCE CHARACTERISTICS

A size reducer operating ideally would have the following characteristics:

1. Product uniform as to size.
2. Minimum temperature rise during reduction.
3. Minimum power requirement.
4. Trouble free operation.

Considerable investigational work has been done regarding the performance of the various grinders used in agriculture, especially burr and hammer mills. The performance of these devices will be discussed in view of known applicable theory and test results.

6.10. Uniformity of Product. The burr mill is believed to produce a more uniform product than the hammer mill. If the hammer mill reduces by impact there are two factors that could contribute to this assumed undesirable feature.

1. Any one grain may be hit a number of times before it has an opportunity to pass through the screen. Since the path of travel of a grain through the grinder is random, the number of times a grain is hit varies and as a result the size of the product varies.

2. The energy dissipated upon contact between a grain and a hammer varies as the square of the velocity. Since the velocity or peripheral speed varies as the rotor radius, then the energy of impact varies as the square of the radius, or the energy of impact at the end of a hammer is four times as great as at a point half-

way between the end of the hammer and the center of the shaft. Consequently a grain that hits near the end of a hammer is more finely divided than one hitting closer to the shaft.

Now consider a number of grains passing between the burrs of a burr mill. Although shear is assumed to be the reduction process, it is probable that most of the reduction is by crushing, particularly if the rate of feed is high. Any one grain is reduced a number of times during the process, the number depending upon the type of burrs being used and the random path of the grain. If the material being ground is brittle and shatters upon fracture, it is possible that the resulting product will be made up of more fine material than desired.

An average sieve analysis by Silver [15] taken from a number of burr and hammer mills of various types shows no significant difference in uniformity between the mills.

The material in this case was of approximately 11 per cent moisture content, which is moderately dry. Wetter material might perform in a different manner. With this exception the reported studies use modulus index as a measure of performance. Since the modulus does not indicate the size distribution in the sample, insufficient data are available to substantiate or refute this accepted performance feature.

6.11. Power Requirements. The exact power required for a specific job is difficult to determine. Type of material, moisture content, fineness of grinding, rate of feed, type and condition of mill, etc., affect the power requirements.

Some power observations by Silver [15] shown in Fig. 6.10 indicate that the more fibrous materials such as barley require more power than a crystalline material such as corn.

Moist grains are more difficult to grind than dry grain. The effect of moisture upon the power requirement is demonstrated in Fig. 6.11 taken from Silver's investigation.

Test data for a hammer mill by Martin and Roberts [11] given in Fig. 6.12 show the important characteristics of a hammer mill using a fan for elevating the ground material.

The power required to operate the mill empty increases swiftly as the speed increases since the power required to operate the elevating fan varies as the cube of the speed (sect. 5.17). Consequently, eight times as much power would be required to operate

the fan at 4400 rpm as at 2200. The empty horsepower requirement of a mill without a fan is a straight-line function. Note that the useful power, the difference between horsepower consumed and horsepower for running the mill empty, is a small percentage of the input. From the standpoint of power consump-

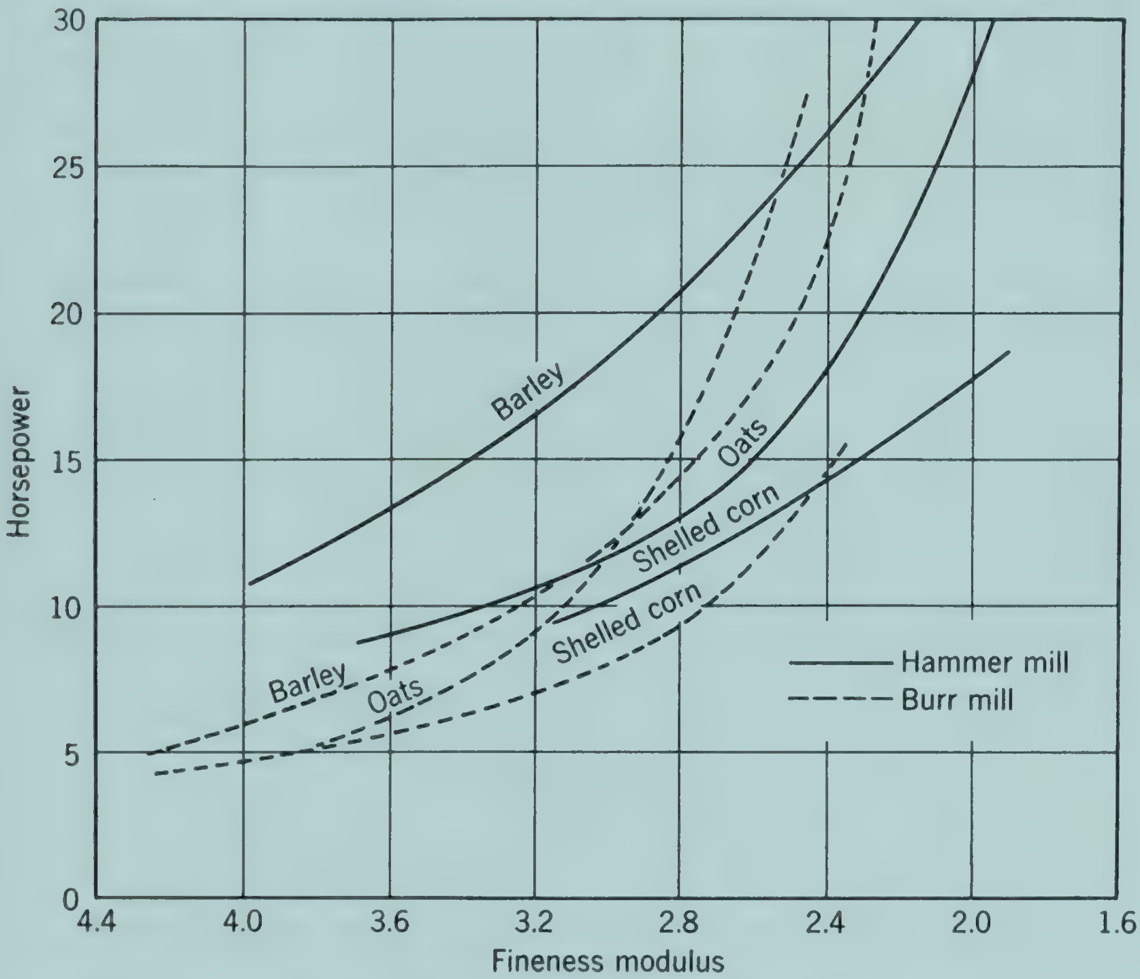


Fig. 6.10. Relationship between power and fineness of grinding for three grains as observed by Silver.[15] Mills are operated at rated speed. Grinding rate is 40 bu per hr. Coarse, medium, and fine burrs are used.

tion, operation below the rated speed is more advisable than operation above.

Capacity in this case is limited by the power available. In order to maintain speeds above 3600 rpm, the feeding rate has to be reduced so that a larger portion of the available power can be used for maintaining mill speed.

6.12. Temperature Rise. The energy for grinding feed is dissipated as heat energy and raises the temperature of the ground product, the mill, and the ambient air. Some heat energy is lost in vaporizing moisture.

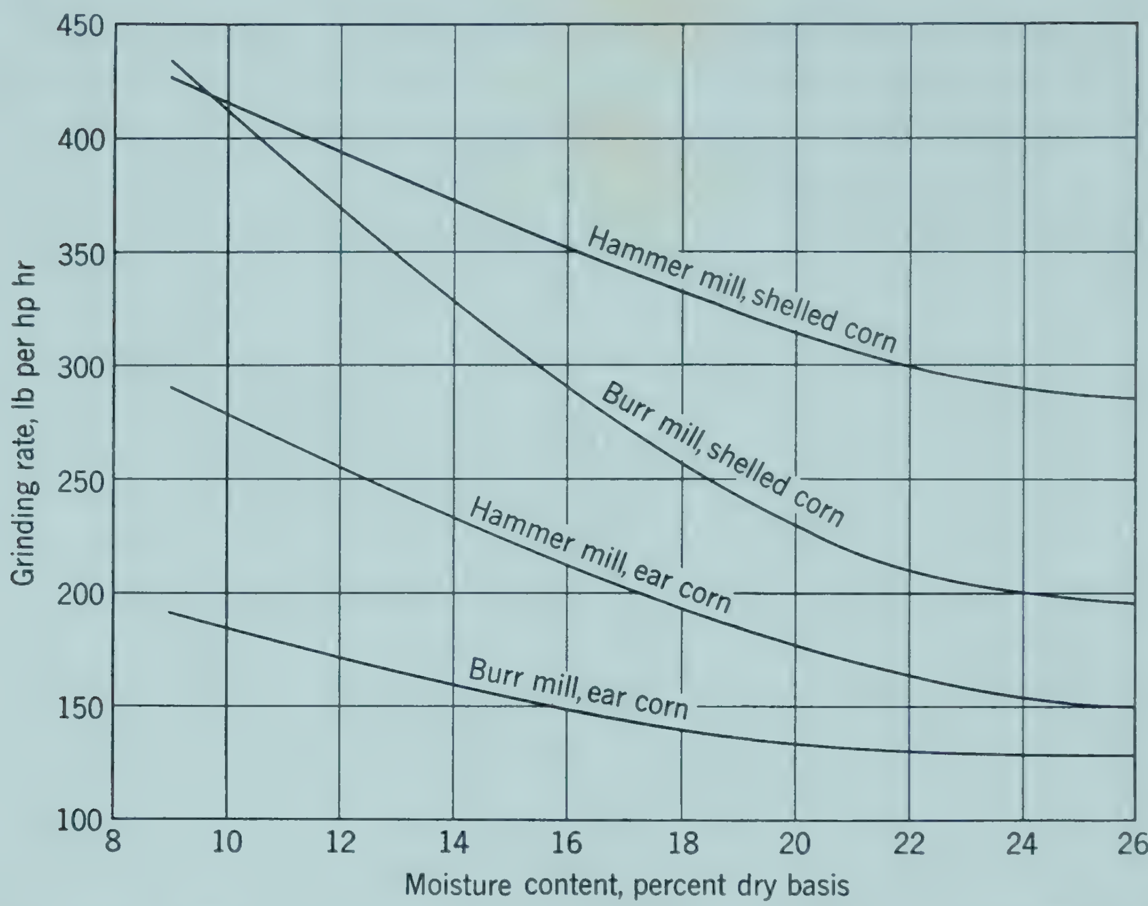


Fig. 6.11. Effect of moisture content on capacity as observed by Silver.[15]

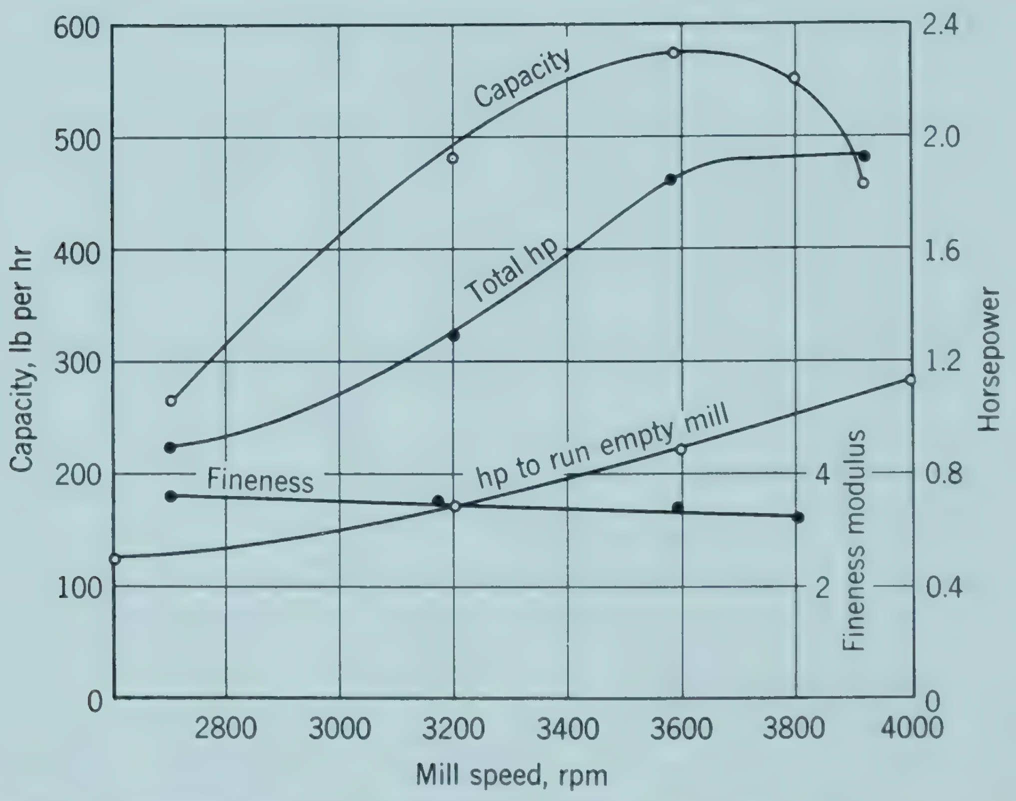


Fig. 6.12. Performance of a hammer mill with a fan for removing the ground material grinding shelled corn. (By Martin and Roberts.[11])

The temperature may rise 50°F or more when grinding fibrous materials such as oats or ear corn in a burr mill, particularly if a relatively fine grind is being produced. Temperature elevations of some materials observed by Silver [15] are shown in Table 6.3.

Table 6.3 MATERIAL TEMPERATURE RISE DURING GRINDING

Material	*Type of Grinder*	*Fineness Modulus*	*Temp. Elevation, °F*
Oats	Burr	2.73	50
	Hammer	2.70	18
Barley	Burr	3.66	8
	Hammer	3.66	2
Shelled corn	Burr	3.96	7
	Hammer	3.13	10
Ear corn	Burr	3.07	14
	Hammer	3.05	13

The hammer mill produces a cooler product because of the large amount of air circulated with the ground grain. High temperatures contribute to decomposition of the ground material, especially if the moisture content is high.

MIXING

Mixing or blending of ingredients for animal feeds, fertilizers, and seed stocks is an extensive processing operation.

Difficulty in mixing may result if the solids are the same size and shape but of different specific gravity, or if they are of different size or shape. Heavier particles tend to remain near the bottom of the container during a mixing operation. Round or small particles tend toward the top. This tendency can be overcome by lifting the materials, more or less in mass, from the bottom of the mixing container and turning them onto and with the top portion. A satisfactory mixing process (1) produces a uniform mixture, (2) in a minimum time, (3) with a minimum cost for overhead, power, and labor. A discussion of the procedures follows.

6.13. Batch Mixers. Batch mixing is used for moderate to small operations where overhead costs must be low and labor costs are not critical. The ingredients may be weighed or measured.

A rotating drum, box, or barrel perhaps with nonsymmetrically located supports is satisfactory for small operations. Flights to assist in lifting the material are recommended. A stationary con-

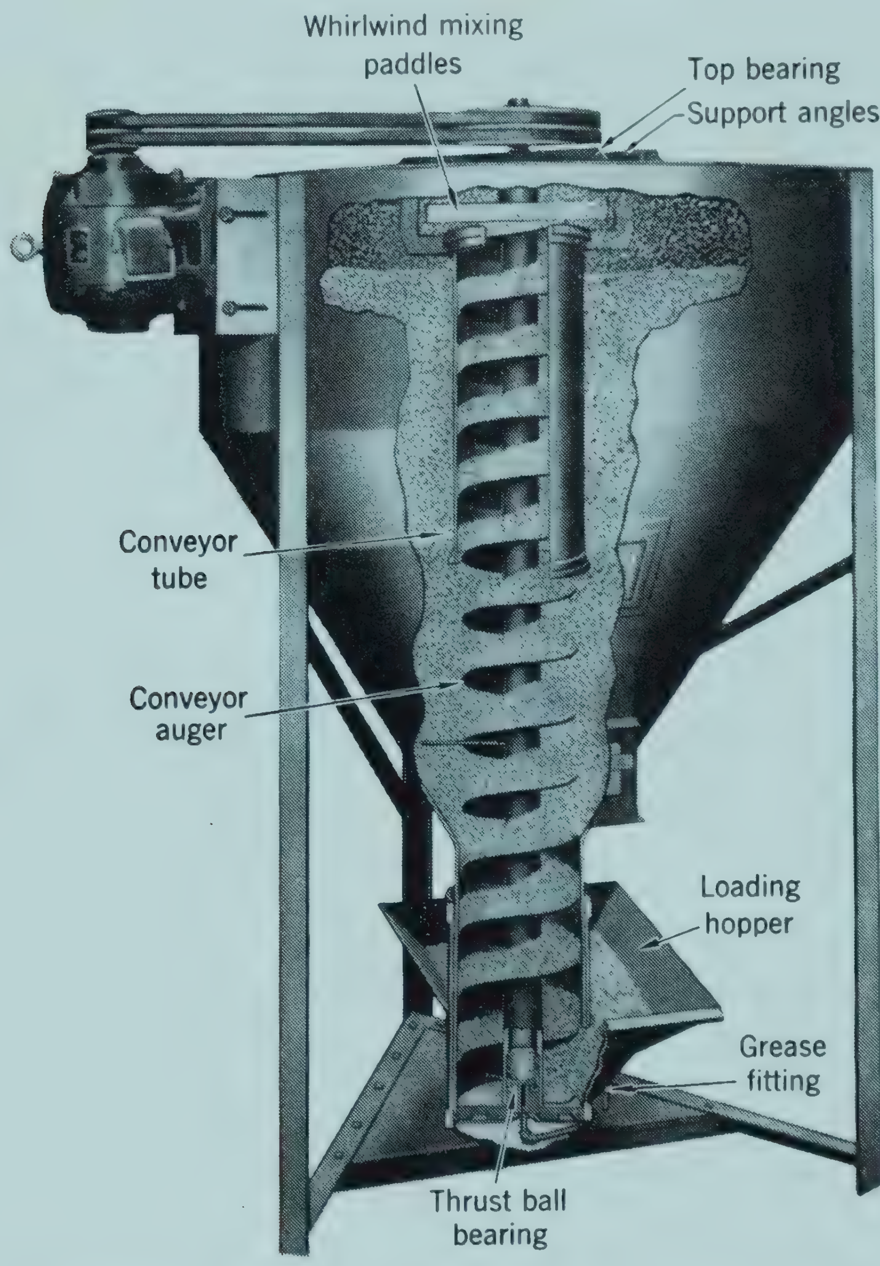


Fig. 6.13. Phantom illustration of a batch type mixer used for mixing livestock feeds. (*Courtesy* Brower Manufacturing Co.)

tainer, usually U-shaped, with rotating paddles or ribbons is used for larger or more difficult mixing operations. The auger system shown in Fig. 6.13 is applied extensively.

The batch method can be adapted to a semicontinuous process by using a number of batch mixers which empty into a common conveyor or storage. Concurrent filling, mixing, and emptying facilitate the use of labor and equipment.

6.14. Continuous Mixers. Continuous mixing procedures are most satisfactory for large, extensive operations. The ingredients are usually added volumetrically by an auger, star wheel, or other device to a screw conveyor. If more accurate control is required, automatic weighing machines may be used.

The continuous mixing operation is carried out in a screw conveyor that may have special flights to insure a thorough mixing job. If the blended product is conveyed some distance, no special mixing unit may be required since the conveying operation will mix satisfactorily.

6.15. Seed Treating. The application of fungicides and insecticides to control fungi and insects that attack seed before and after planting is known as seed treating. The process consists of mixing certain chemicals with the seed. Since the rates of application are low, in the order of ½ to 2 oz per 100 lb, mixing must be thorough to insure satisfactory results.

Mixing procedures are comparable to those discussed in the previous sections. Since the quantities processed are relatively small when compared to feed-mixing operations, batch processes are usually used. Most of the dusts used are toxic, therefore special health precautions must be observed.

REFERENCES

1. Alden, J. L. *Design of Industrial Exhaust Systems.* The Industrial Press. 1939.
2. Bohstedt, G. Feed Processing in Relation to Animal Nutrition. *Agr. Eng.* 17:98–100. 1936.
3. Bruhn, H. D. Burr Mill Design and Performance. *Agr. Eng.* 17:101–106. 1936.
4. Colby, H. N. Feed Grinding, Mixing, and Elevating. *Wash. Agr. Expt. Sta. Popular Bull.* 151. 1937.
5. Duffee, F. W. Testing the Efficiency of Silage Cutters. *Agr. Eng.* 5:3–6. Jan., 1924.
6. Duffee, F. W. Efficiently Filling the Silo. *Agr. Eng.* 6:4–12. Jan., 1925.
7. Duffee, F. W. A Study of Factors Involved in Ensilage Cutter Design. *Agr. Eng.* 7:84–87. Mar., 1926.
8. Fenton, F. C., and Logan, C. A. Farm Grinding of Grain and Forage. *Kans. Eng. Expt. Sta. Bull.* 27. 1931.
9. Kable, G. W. Results of Research in Feed Grinding. *Agr. Eng.* 8:53–54. 1927.
10. Lapple, C. E., and Shepherd, C. B. Calculations of Particle Trajectories. *Ind. Eng. Chem.* 32:605–617. 1940.

11. Martin, J. W., and Roberts, J. Feed Grinding with Small Electric Motors. *Kans. Agr. Extension Bull.* 87. 1941.
12. Morrison, F. B. *Feeds and Feeding.* Abridged. Morrison. 1937.
13. Nicholas, J. E. Performance of Small Hammer Mills. *Agr. Eng.* 21: 207–210. 1940.
14. Russell, H. G. The Why and When of Feed Grinding. *Ill. Extension Service Mimeo Circ.* Pp. 1–5. 1940.
15. Silver, E. A. Feed Grinder Investigations. *Ohio Agr. Expt. Sta. Bull.* 490. 1931.

PROBLEMS

1. The average minimum or representative dimension of the material retained on a sieve in the set used for a fineness modulus determination is approximately 1.4 times the sieve opening. Why?
2. Derive equation 6.1.
3. Prove that the fineness modulus is a geometric mean of a minimum dimension weighted on the basis of quantities, that is,

$$\log_2 D_{\text{av}} = \frac{\Sigma \log_2 D}{\Sigma w}$$

where w is the weight or per cent of material of minimum dimension D.

4. Determine the fineness modulus, uniformity index, and average particle size for the following sieve analysis:

Mesh, in.	*Per Cent Retained*
$\frac{3}{8}$	2
4	6
8	10
14	40
28	28
48	12
100	2
Pan	0

5. Assume the particles in problem 2 to be spherical and the result of a reduction from a uniform product ⅜ in. in diameter. Prepare and complete a table with the following column headings: (1) sieve mesh, (2) sieve-opening width, (3) relative number of particles retained by each sieve, using 2 on the ⅜-in. sieve as the base, (4) total relative area represented on each sieve. Discuss the data from the standpoint of (1) power requirements, (2) rate of oxidation of air sensitive materials, (3) importance of a minimum amount of fine material.
6. Determine the constants of equation 6.7 for shelled corn and barley ground by hammer mill from the curves of Fig. 6.11. Express an opinion as to the deviation from Kick's and Rittinger's laws.
7. If all the grinding power, Fig. 6.12, at 3400 rpm is accumulated as heat in the ground material, what is the temperature rise of the material? Assume the specific heat to be 0.30.

CHAPTER 7

Cleaning and Sorting

NOMENCLATURE

A = projected area of particle, sq ft.
a = accelerational force, ft per sec^2.
b = entry width, ft.
C = particle aerodynamic-drag coefficient, dimensionless.
C_f = centrifugal force, lb.
D = average particle diameter, ft.
d = diameter, ft.
d' = particle diameter, in.
E = exit duct diameter, ft.
F = force, lb.
g = acceleration of gravity, 32.2 ft per sec^2.
H = cone height, ft.
h = entry height, ft.
K = vane constant, dimensionless.
k = a constant, dimensionless.
L = length, ft.
M = particle mass, lb per sec^2 per ft.
N = number of turns.
n = revolutions per minute.
P = pressure drop, number of velocity heads.
R = radius of rotation, ft.
Re = Reynolds number.
r = radius, ft.
S = separating coefficient, dimensionless.
V = relative velocity, ft per sec.
v_p = particle volume, cu ft.
W = particle weight, lb.
γ = fluid specific weight, lb per cu ft.
γ_p = particle specific weight, lb per cu ft.
μ = viscosity, lb per ft-sec.
θ = time, sec.

The comparative commercial value of a farm product is a function of its grade factor. Grade factors have been established by various agencies interested in particular products and are recog-

nized by these agencies as the legal standard for commercial use. Meat, dairy products, fruits, vegetables, forage crops, fiber crops, grains, tobacco, etc., and their products are graded on the basis of standards that are available, in general, from the Office of Marketing Services, U. S. Department of Agriculture, Washington, D. C.

7.1. Grade Factors. Grade factors that apply in various combinations to all the products produced on the farm could be classified thus:

1. Physical characteristics.
   a. Moisture content.
   b. Unit size.
   c. Unit weight.
   d. Texture.
   e. Color.
   f. Foreign matter.
   g. Shape.
2. Chemical characteristics.
   a. Analysis (composition).
   b. Rancidity, free fatty acid index (for fat-containing material).
   c. Odor and flavor.
3. Biological.
   a. Germination.
   b. Type and amount of insect damage.
   c. Type and amount of mold damage.
   d. Bacteria count.

A processing aim is to handle and manipulate products so that they will yield the highest possible net return after being processed, initial quality of the raw products being recognized as the important prime consideration.

General procedures that may be used to improve, maintain, or change the quality of a product are:

1. Control storage conditions which are:
   a. Temperature.
   b. Relative humidity.
   c. Time.

2. Kill or inhibit destructive organisms by:
   a. Fumigation.
   b. Refrigeration.
   c. Heating.
3. Improve the physical characteristics by:
   a. Changing or maintaining the moisture content.
   b. Removal of foreign or dissimilar material.
   c. Sorting into various fractions.

The procedures listed above have or will be treated directly or indirectly in the chapters on air-vapor mixtures, refrigeration, and drying except 3*b* and *c*, removal of foreign or dissimilar material and sorting into various fractions which would be cleaning and sorting and would apply to preparation of a material for processing as well as grading of a commercial product.

Cleaning refers to the removal of foreign or dissimilar material. This may be done by washing, screening, hand picking, or by other means which are described later in this chapter.

Sorting refers to the separation of the cleaned product into various quality fractions that may be defined on the basis of size, shape, density, texture, and color. A distinction should be noted between sorting and grading. Grading implies the classification of the material on the basis of commercial value and usage and is dependent upon more factors than are recognized when physical sorting is considered. For example, small grain and certain fruits and vegetables may be passed through a mechanical "grader" which sorts the material on the basis of size, shape, or density. The mechanical "grader" is not effective as regards moisture content of the material, the amount of fungal or insect damage, or germination. Consequently, the resulting fractions are not *grades* as usually recognized. On the other hand, if the material had previously been sorted on the basis of moisture content, damage, etc., the fractions produced mechanically might have been commercial grades.

The unit operations used in cleaning and sorting cannot be listed exclusively under either of these heads since some operations apply to both. For example, an air blast can be used for cleaning chaff from grain or for removing lightweight seeds from seed stock. A sieve can be used for cleaning sticks and leaves from grain or for sorting out small, immature grain. The unit-

operation discussion that follows does not recognize the cleaning and sorting classification as such.

WASHING

Fruits, vegetables, and nuts are sometimes washed to remove dirt, spray residues, and other foreign material. A preliminary cleaning operation by rough screening may be used in certain instances; but the product may be bruised, and therefore usual practice is to place the product as delivered directly into the washer.

Washers may be continuous or batch type. The batch type of washer is recommended only for small plants or institutional or community installations where use is intermittent and operating costs are not critical. Washing is performed by one or a combination of the following:

1. Soaking in still or moving water or other fluids.
2. Water sprays.
3. Rotary drum.
4. Rotating brushes.
5. Shuffle or shaker washers.

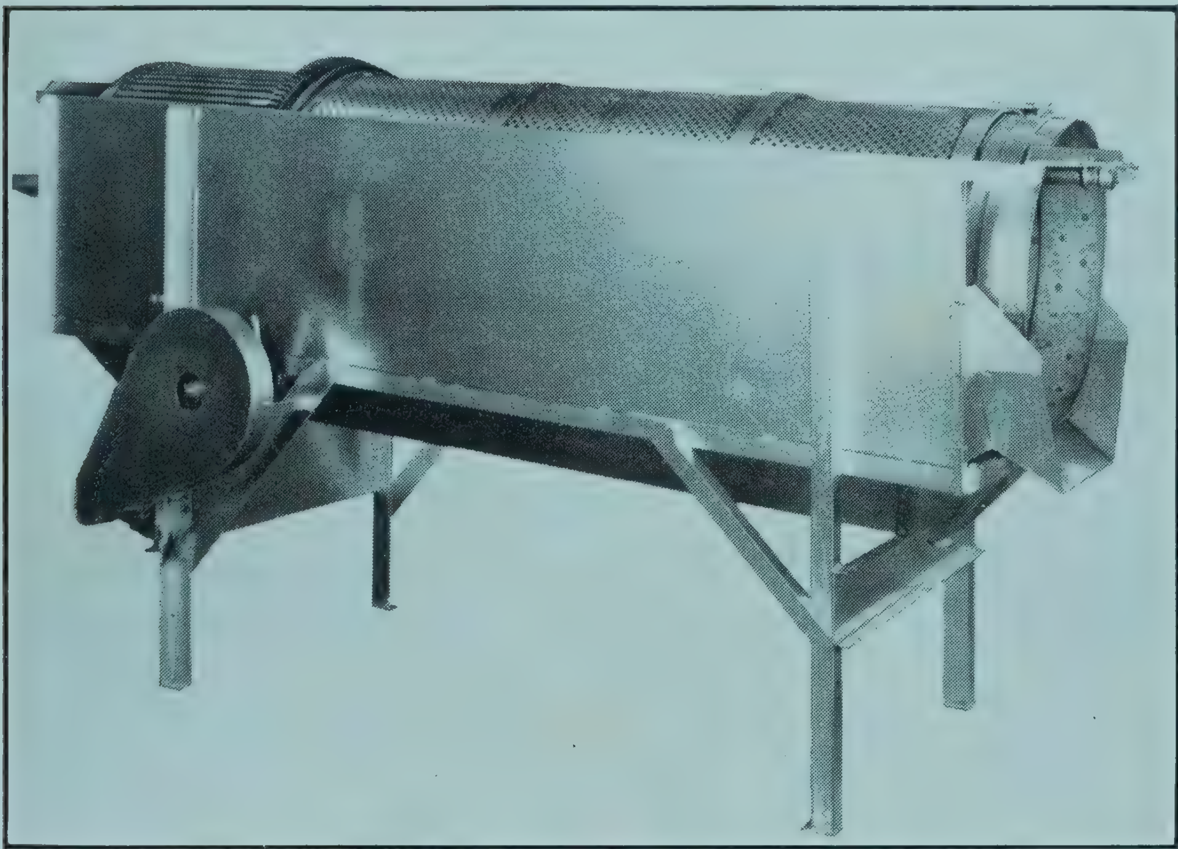
7.2. Soaking. Soaking in still or moving water or other fluids is effective only if dirt, or other surface undesirables, is present in small quantities and is loosely attached to the product. This method is frequently used in connection with other methods as a precleaner or soaker.

7.3. Water Sprays. Water sprays that vary from low-pressure wide-angle to very high-pressure directed jets are very effective since they physically remove firmly attached pieces of dry dirt and agitate the mass of product, particularly if it is carried in a water bath. Sprays are suitable for most products, but the intensity and type of spray distribution must be carefully selected. It is evident that a high-pressure concentrated spray for cleaning potatoes would be destructive to celery or lettuce.

Flood washing is done by a large quantity of water moving at a moderate to high speed over the product.

7.4. Rotary Drum. This washer is the most commonly used commercial washing device because of simplicity, high capacity, thorough cleaning, and a minimum of damage to the product. It

may be used in a bath of water or with spray nozzles, or both. A rotary-type washer is shown in Fig. 7.1. The performance as based upon dirt removal is dependent upon the rotative speed, the roughness or amount of corrugating on the inside surface, and the



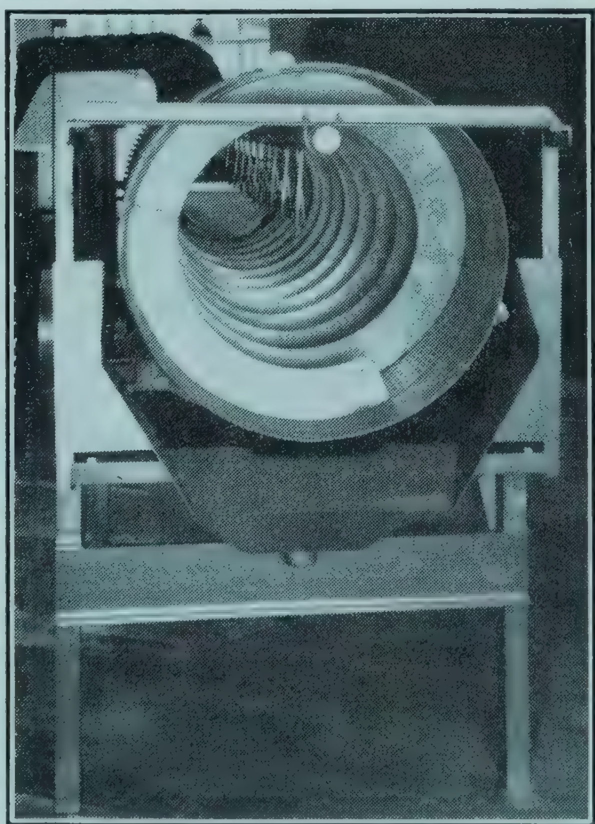


Fig. 7.1. A rotary washer. The material to be washed is screwed through the rotating drum. Note the nozzles for the application of the water. (*Courtesy* Food Machinery Corp.)

length of time the product is retained in the washer. Auxiliary aids such as spray nozzles may be used to facilitate the job. The washing time is controlled by the pitch of the drum or by helical retaining fences which "screw" the product through the drum.

7.5. Brush Washers. Rotating brushes are frequently used and are highly effective. They are particularly effective for removing sandy or loamy soil and for removing spray residues. Washing time is controlled by the relative motion of the brushes which moves the product through a definite path and to some extent by the flow of wash water if the brushes are operating in a water bath. The brushes are made of fiber, rubber, sponge, or other material and may have to be replaced frequently. This expense must be considered in evaluating this type of washer.

7.6. Shuffle or Shaker Washers. These washers employ a vigorous reciprocating motion. Since the action is reciprocating, the washer must be ruggedly constructed and carefully maintained to minimize interruptions resulting from mechanical failure. Although this type of washer is more complicated mechanically and more expensive than some of the other types, it is to be recommended for the more difficult cleaning jobs. Although effective because of its vigorous action, it is obviously unsuited for products that are easily damaged.

A sorting screen is frequently included in the unit so that dirt, pieces of leaves, stems, and product fragments are washed away from the material.

The best washing procedure usually utilizes two or more of the washing devices described. Also, the washing procedure can usually be integrated with the movement of the raw product into the plant. For example, washing and elevating can be combined. Water flowing in a baffled trough will soak and remove part of the dirt and at the same time move the product from one operation to another. Spray nozzles used in connection with the other washing devices frequently improve the performance.

Spray residues are removed from fruit by washing with a detergent solution, with a 0.5 to 1.0 per cent hydrochloric acid solution at 70°F, or both. Immersion in a vat for periods up to 3 min or power-spray washing for smaller periods will remove the average deposit. Heavy wax formations or heavy oil-spray residues may require higher water temperatures or alkaline washes. However, since these may damage the fruit, it is recommended that they be used only in extreme cases.

Chemicals are frequently incorporated in the wash water to control fungus, insect, and bacteria growth. The laws of the Federal Food and Drug Administration limit the sterilants used and their residual tolerance where the products are involved in interstate commerce. Most states have comparable laws pertaining to products that are sold within the state.

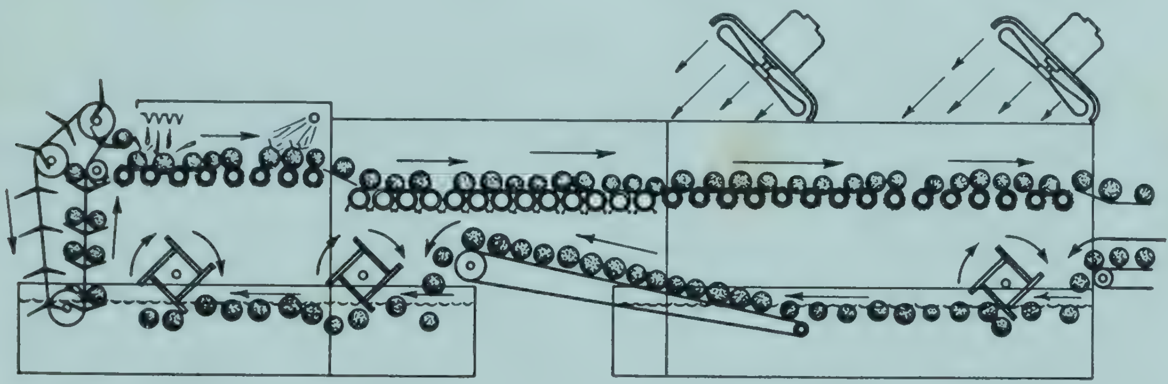


Fig. 7.2. A combination washer showing, in order, two soaking operations, a spray washing operation, water removal, and drying. The soaking bath may contain sterilizing chemicals, a material for spray residue removal, or may be compounded for other purposes. The last few rows of brushes may be used for waxing.

A commercial washer using a number of the procedures is shown schematically in Fig. 7.2. The brushes may be used for waxing when advisable.

SORTING FRUITS AND VEGETABLES

Fruits and vegetables are sorted on the basis of color, damage, and size. Most sorting on the basis of color and damage is done manually, but the electric eye has been used successfully in pilot studies, and its future general application appears promising.

Most fruits and vegetables are subject to damage if handled too vigorously. Since a delicate contact with the material and high capacity are necessary, some unique procedures have been devised.

7.7. Screens.* Many fruits and spherical vegetables are graded over vibrating screens made of copper, stainless steel, or other materials which do not react chemically with the products.

The unsized material passes onto a vibrating or rotating screen that is perforated to pass all but the largest material, which goes

* Note sect. 7.11 for a more complete discussion of screens as used on small grain.

over the end of the screen and thence to the canning or other processing operation. The material that passes the top screen is rescreened, and the new fractions are moved to other operations. The characteristics of the vibrating motion if used must be carefully controlled in order to minimize damage to the material. A perforated rotary drum is sometimes used for peas and similar symmetrical firm products.

The screens used for fruit and vegetable sorting or grading are furnished in $\frac{1}{32}$-in. increments. An example of the usual sizes recognized for various grades of a few canned fruits are shown in Table 7.1.

Table 7.1 AVERAGE MINIMUM DIAMETER OF VARIOUS GRADES OF CANNED FRUITS

(After Cruess and Christie's *Laboratory Manual of Fruit and Vegetable Products*)

	Fancy	*Choice*	*Standard*
Apricots	$\frac{56}{32}$ in.	$\frac{54}{32}$ in.	$\frac{50}{32}$ in.
Cherries, Royal Anne	$\frac{24}{32}$ in.	$\frac{28}{32}$ in.	$\frac{28}{32}$ in.
Cherries, black	$\frac{26}{32}$ in.	$\frac{25}{32}$ in.	$\frac{22}{32}$ in.
Grapes, muscat	$\frac{26}{32}$ in.	$\frac{25}{32}$ in.	$\frac{24}{32}$ in.
Peaches	$\frac{76}{32}$ in.	$\frac{64}{32}$ in.	$\frac{56}{32}$ in.
Plums, green gage	$\frac{56}{32}$ in.	$\frac{50}{32}$ in.	$\frac{42}{32}$ in.
Pears, Bartlett *	8–10 pieces	10–12 pieces	15–17 pieces

* Pieces per No. $2\frac{1}{2}$ can.

It must be recognized that damage to the product will result if this type of sorter is not properly operated or if overloaded.

7.8. Diverging Belts. A widely used grader consists of two belts which diverge as they travel. The fruit is carried on and between the belts. Since the distance between the belts increases systematically, the smaller pieces will drop between the belts at the beginning of travel whereas the larger pieces will be carried farther.

7.9. Roller Sorters. Roller sorters are fast, accurate, and cause little damage to the fruit. The type of roller sorter shown in Fig. 7.3 is used extensively in the fruit industry. Each roll rotates in a counterclockwise direction. The fruit is continuously rotated so that each individual piece has an opportunity to register its minimum dimension with the space in the sorter.

This roller sorter is divided into three roll units, which are hinged so that the gaging space increases progressively through the sorter. When the rolling unit closes the space at the end of its travel, the turning roller ejects without damage any fruit trying to pass through the space.

Roller conveyors with fixed space between the rolls are used for removing small fruit, twigs, and leaves.

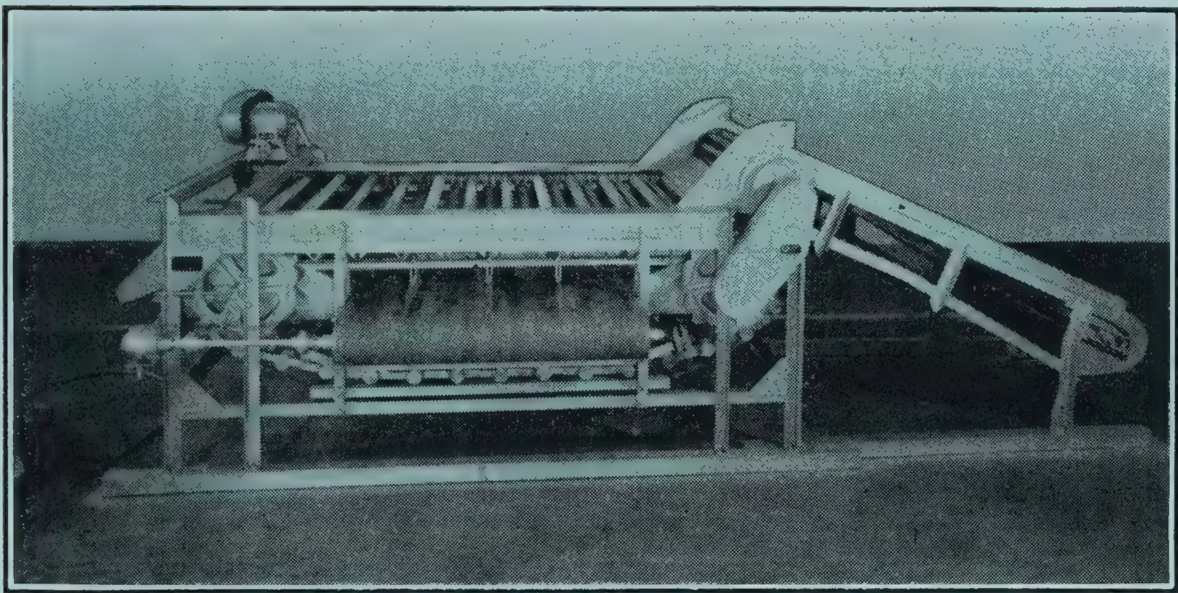


Fig. 7.3. A roller sorter for citrus fruit. (*Courtesy* Food Machinery Co.)

7.10. Weight Sorters. Sorters that operate on the basis of unit weight are accurate and moderately fast and damage to the material is minimized. Weight sorters can be used on all large-size products, apples, oranges, canteloupe, for example, and are adapted to egg handling. They are especially useful for sorting materials that because of shape or texture are not adapted to other procedures.

The material is placed into individual cups by an automatic indexing feed. As the cups travel through the sorter they are indexed with spring-loaded trips. The spring tension is progressively weaker from beginning to end of movement. The heavier units overcome the spring reaction and are discharged at the beginning of travel, the lighter units moving a greater distance before being discharged.

This sorting principle is not limited by size or shape of material. The fineness of separation is dependent upon precision of design and calibration accuracy. It will not handle as large a quantity of material per unit of time as other sorters.

CLEANING AND SORTING GRAIN, NUTS, AND SEED

No distinct division can be made between cleaning and sorting of grain and various seed stocks since the process is carried on simultaneously and the procedures are common to both.

Cleaning, sorting, and partial or perhaps final grading or classifying of the products being considered are based upon the following characteristics of the material:

1. Size.
2. Shape.
3. Specific gravity.
4. Surface characteristics.

The first three are the most important. Surface characteristics as differentiated from shape affect the drag coefficient where an air blast is used for separation. Although it is known to be an effective factor, its importance thus far has not been demonstrated. Roughness is used in certain difficult cleaning operations, which are discussed in sect. 7.16.

7.11. Screens. The most widely used sorting device is the screen or sieve. Screens used in conjunction with an air blast will satisfactorily clean and sort most granular products. The screening unit is composed of two or more screens as shown in Fig. 7.4. These screens are suspended by hangers in such a way that they have a horizontal oscillating motion H and a smaller vertical motion V, Fig. 7.4.

The combination of these two motions moves the grain down the screen and at the same time tosses it sufficiently so that the sheet of grain is thoroughly stirred. The screen pitch is adjustable. This controls the rate of downward travel of the grain. Screens are generally available with round, triangular or slotted holes. Slotted screens may be punched sheet metal or wire cloth. The slots may be orientated in the direction of travel, perpendicular to it, or both.

Performance is based mainly upon careful selection of the screens, although screen pitch is important. Generally speaking, a relatively steep pitch does a better job. The material flows at a faster rate and, consequently, is not so deep. On the other hand,

it must not flow so fast that there is insufficient time for each individual grain to register with a hole.

No two cleaning and sorting jobs are the same. Screen selection and adjustment must be carefully made if a fine and complete separation is to be assured. Although experience in this regard is an asset, certain principles will help materially in setting up and operating the screens.

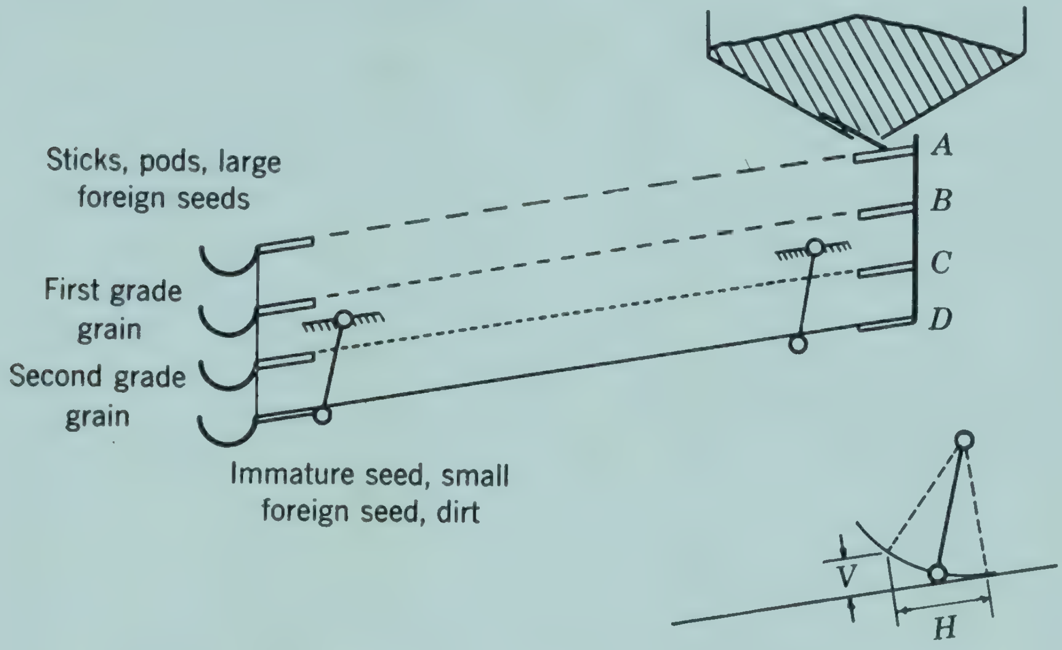


Fig. 7.4. Basic screen arrangement for sorting grain.

If the material to be cleaned and sorted is generally spherical, round-hole screens are recommended. Screens with oblong holes should be used if the material varies systematically from the spherical and has a uniform minimum dimension characteristic. Oats, alfalfa, flax, corn, and pumpkin seed would fall in the oblong class. Beans, wheat, onion seed, and sorghum seed would be considered in the round class.

The top sieve A, in Fig. 7.4, is called the scalper and is for scalping off material larger than that to be retained. It might be considered as a cleaner since it sorts out the undesirable dissimilar material larger than that to be retained. Generally, the top screen A should have round holes that should be just large enough to pass the material to be retained. Although round-hole screens do a better job of rejecting small sticks, chaff, leaf parts, etc., they should not be used if the material is flat and adapts itself best to the oblong hole.

The second screen B should have holes just small enough to retain the best material. If the material is basically spherical and the top screen has round holes, the second screen B will probably perform best if it has slotted holes. Conversely, if the top screen has slotted holes, the B screen will probably perform best with round holes.

Frequently, screen C is omitted, the light seed and small weed seed being permitted to fall onto the solid pan D. On the other hand if the material is to be sorted into additional quality groups

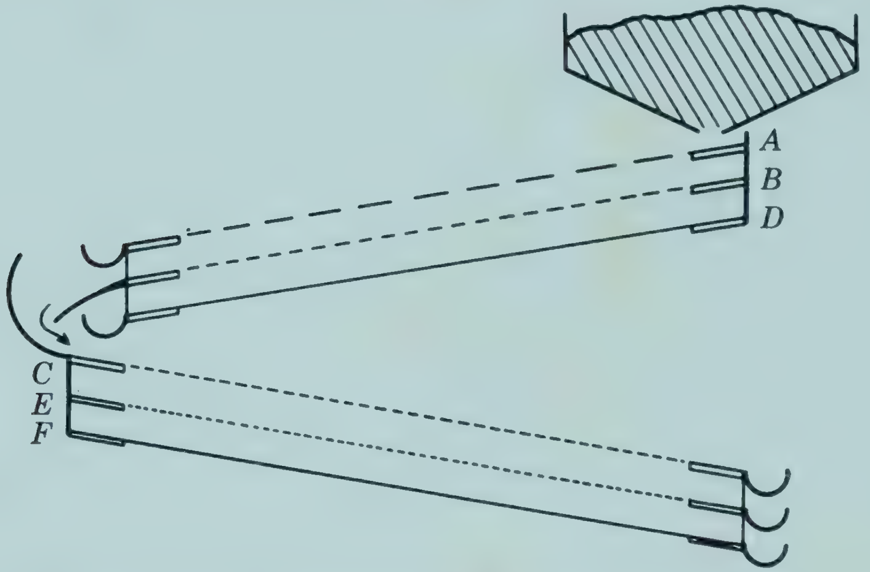


Fig. 7.5. Efficient screen arrangement for most cleaning and sorting jobs.

or if it is a mixture of materials that are to be sorted and retained, one or more additional screens may be required.

A more efficient screen arrangement is shown in Fig. 7.5. A rough but accurate separation is made by the upper set of screens. The mixture that is to be sorted is freed of both large and small undesirable material and is conducted to the lower set of screens where a finer separation is made. This arrangement is more efficient than that shown in Fig. 7.4 since (1) the material is not contaminated with the small fractions rejected by D, (2) the pitch of the lower bank of screens can be varied relative to the upper bank, and (3) the vibrating characteristic can be varied between the two sets to accommodate the job at hand.

When a fine degree of sorting is being made, grains will frequently lodge in the holes. Screens are frequently fitted with a brush which travels under the screen and pushes the lodged material back through the screen. There are other equally effective devices.

Screen cleaners that employ an air blast to assist in cleaning are generally known as fanning mills. A conventional fanning mill of the type used by seed houses is shown in Fig. 7.6. Although they can be and are used for most cleaning and sorting work, they are not as selective as other devices in regard to

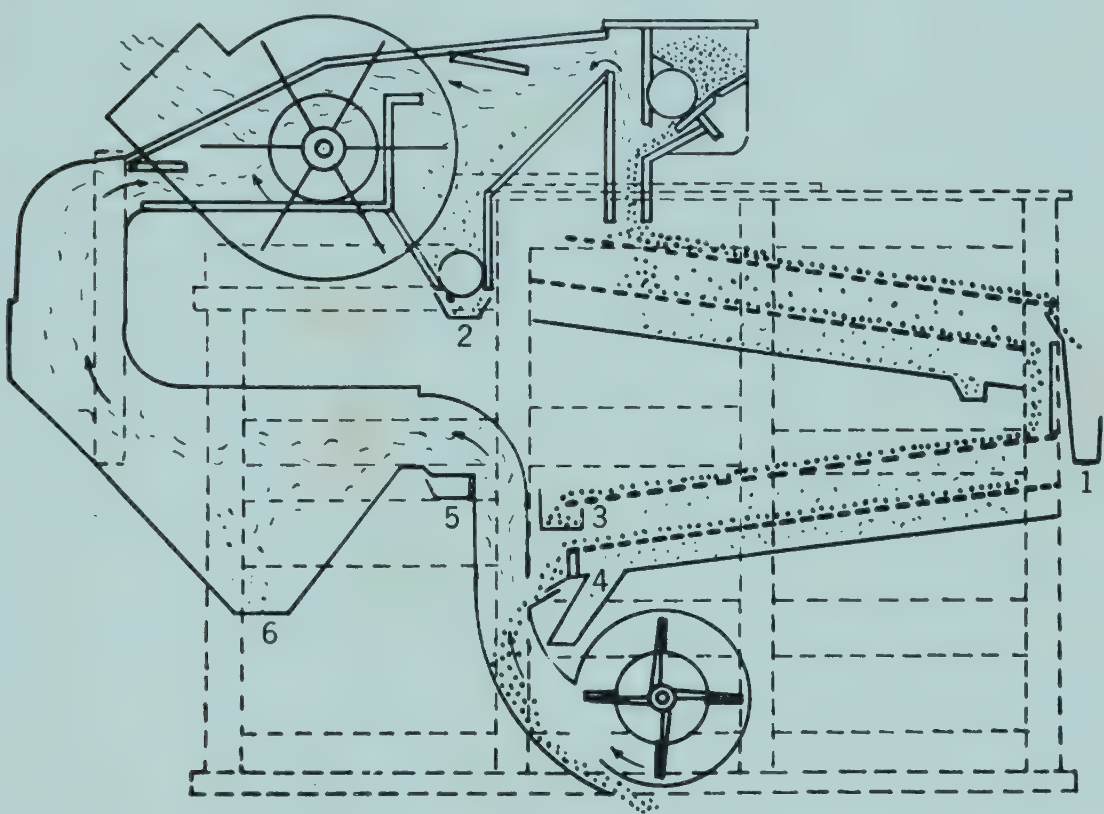
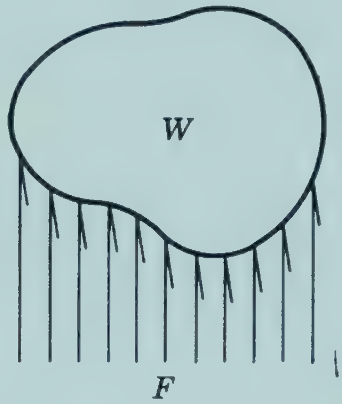


Fig. 7.6. A conventional fanning mill shown in cross section. The rough mixture is fed from the hopper at the top. Large-sized foreign material is taken off at 1. Dust and very light seeds are aspirated at the hopper, the light seeds accumulating at 2. The desirable seed accumulates at 3 and discharges at the bottom, the smallest seed accumulating at 4. Fractions between 3 and the bottom discharge are deposited at 5 and 6. (*Courtesy* A. T. Ferrell and Co.)

density. Also, if the seeds to be sorted have the same or nearly the same shape or dimension, some other device is necessary.

7.12. Aerodynamics of Small Particles. A particle in free fall will reach a steady-state velocity that depends upon the physical characteristics of the particle, the fluid in which it is falling, and the accelerational force. This particle characteristic is useful in pneumatic separation and conveying. The steady-state velocity is also the air (or liquid) velocity required to suspend or balance a particle, thus the applicability to processing operations.

The following analytical procedure is adapted from a treatment of this particle characteristic by Lapple and Shepherd.[5]



The forces involved by a particle falling are

$$M(dV/d\theta) = (v_p\gamma_p - v_p\gamma) - F$$

$$= Mg\frac{\gamma_p - \gamma}{\gamma_p} - F \tag{7.1}$$

By definition

$$F = C(V^2/2g)\gamma A \tag{7.2}$$

and equation 7.1 can become

$$dV = \pm g\frac{\gamma_p - \gamma}{\gamma_p} - C\frac{V^2\gamma A}{2gM} \tag{7.3}$$

where A = projected area of particle, sq ft.
γ = fluid specific weight, lb per cu ft.
γ_p = particle specific weight, lb per cu ft.
v_p = particle volume, cu ft.
C = particle aerodynamic drag coefficient, dimensionless.
V = relative velocity, ft per sec.
F = force, lb.
M = particle mass.
W = particle weight, lb.
θ = time, sec.
Re = Reynolds number.
D = average particle diameter, ft.

The sign of the g (gravity) term is positive for a particle starting from rest or having an initial downward velocity. The sign is negative for an initial upward velocity.

If γ_p is larger than γ the particle motion will be downward when steady-state conditions have been reached. If the fluid is denser than the particle, that is, γ is larger than γ_p, the particle will rise during the steady-state condition.

For constant velocity, steady-state conditions, $dV/d\theta$ is zero and equation 7.3 becomes

$$V = \sqrt{\frac{2g^2M(\gamma_p - \gamma)}{CA\gamma_p\gamma}} \tag{7.4}$$

A direct solution of equation 7.4 is impossible since C is a function of V. The velocity V can be determined explicitly by the following procedure, however.

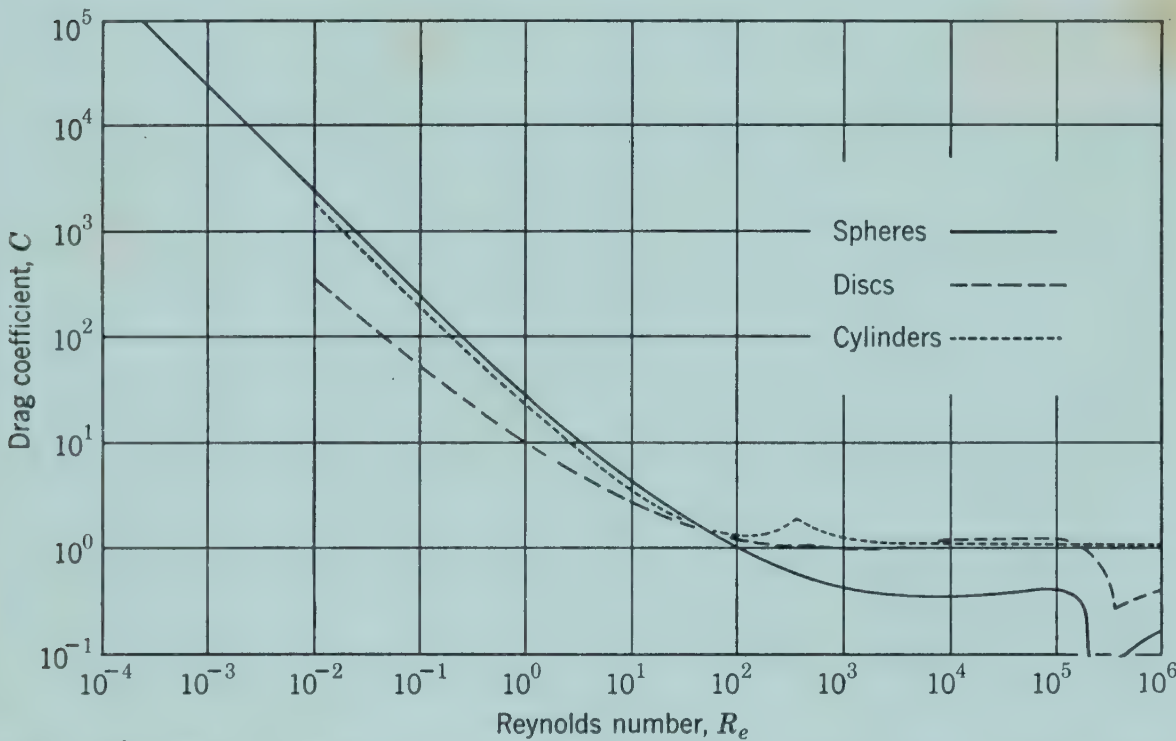


Fig. 7.7. Drag coefficient as a function of Reynolds number (Table 7.2). The axis of the disc is parallel to the fluid motion. The cylinder is of infinite length; its axis is perpendicular to the fluid motion.

Studies with spheres, discs, and cylinders have shown a distinct relationship between the drag coefficient C and Reynolds number, Fig. 7.7. This relationship permits a solution in the following manner.

$$Re = DV\gamma/\mu \tag{7.5}$$

and

$$V = Re\mu/D\gamma \tag{7.6}$$

Equations 7.4 and 7.6 are combined, and the following relationship results.

$$CRe^2 = \frac{2gWD^2\gamma(\gamma_p - \gamma)}{\mu^2 A \gamma_p} \tag{7.7}$$

Since the right-hand terms of equation 7.7 are fixed by the system being considered, a value for CRe^2 can be determined.

Table 7.2 RELATIONSHIPS BETWEEN Re, C, AND CRe^2 FOR SPHERES

Re	C	CRe^2
0.1	240	2.4
0.2	120	4.8
0.3	80	7.2
0.5	49.5	12.4
0.7	36.5	17.9
1.0	26.5	26.5
2	14.4	57.6
3	10.4	93.7
5	6.9	173
7	5.4	265
10	4.1	410
20	2.55	1.02×10^3
30	2.00	1.80
50	1.50	3.75
70	1.27	6.23
100	1.07	10.7
200	0.77	30.8
300	0.65	58.5
500	0.55	138
700	0.50	245
1,000	0.46	460
2,000	0.42	1.68×10^6
3,000	0.40	3.60
5,000	0.385	9.60
7,000	0.390	19.1
10,000	0.405	40.5
20,000	0.45	180
30,000	0.47	426
50,000	0.49	1.23×10^9
70,000	0.50	2.45
100,000	0.48	4.8
200,000	0.42	16.8
300,000	0.20	18.0
400,000	0.084	13.4
600,000	0.100	36.0
1,000,000	0.13	130
3,000,000	0.20	1.8×10^{12}

A plot or table of CRe^2 vs. Re values prepared from Fig. 7.7 for a specific-shaped particle permits the Reynolds number for steady-state conditions to be found. Table 7.2 reports these relationships for spheres. Comparable tables for discs or cylinders could be prepared easily.

The Reynolds number can be substituted in equation 7.6, and the steady-state velocity can be calculated.

The velocity-time-distance relationship for the changing velocity period can be determined only by incremental solution of equation 7.4.

Note that this procedure applies for laminar, turbulent, and mixed flow. Turbulent flow exists for Reynolds numbers greater than 500, laminar flow for those less than 2. Laminar and turbulent flow may exist somewhat within the boundaries of the 2 to 500 Reynolds number bracket, but this is generally recognized as the mixed-flow region.

7.13. Application. Agricultural particles, oats, grass seed, ground materials, dusts, sawdust, for example, do not conform to any of the three geometrics discussed. Most particles do fall between two of the geometrics or an approximate equivalent sphere can be assumed.

The geometric mean of the three axial, significant dimensions is the diameter of an approximate equivalent sphere. This diameter should be used for determining the projected area of a particle moving in the turbulent or near turbulent region. The minimum cross-sectional area should be used for particles moving in the laminar or near laminar region. Sieve-size particles can best be assumed spherical and of such a dimension as the sieve analysis indicates.

7.14. Pneumatic Separators. Fanning mills for farm use or small seed-processing plants consist essentially of a set of screens as described in sect. 7.11 and a fan for moving air through the grain which removes chaff, dirt, and lightweight seed. The separating effect of moving air is used by itself and in connection with other devices.

The moving air for this method of cleaning and separating can be provided by "blowing" through the grain, the air coming from the discharge of a fan. Or it can be provided by "drawing" the air through the grain by connection to the intake of a fan. The latter process is called aspirating.

The process set out in sect. 7.12 can be used for determining the performance of pneumatic separators since the settling velocity V is the factor upon which performance is based.

It is probable that the lighter fractions in wheat, for example, and the heavier fractions in oats cannot be separated by this method. Consequently, pneumatic separators are usually used with other cleaning procedures for separations that are not so similar, or on material that has been precleaned or sized if a separation on a density basis is desired.

This principle may also be used to separate falling grain by altering the trajectory as it falls.

7.15. Specific Gravity Separators. Gravity separators such as shown in Fig. 7.8 are of recent adaptation and can make accurate separations under the more difficult situations.

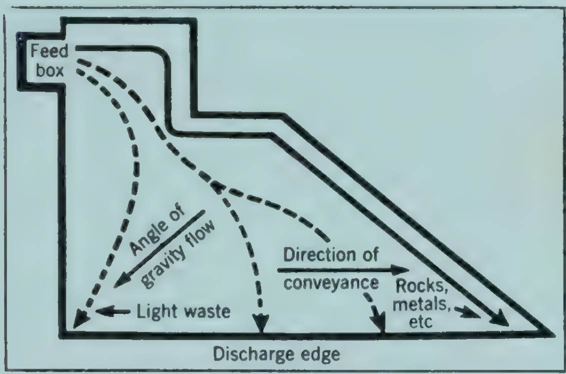
This separating device is based on two conditions: (1) the ability of a grain to flow down an inclined plane and (2) the lifting or floating effect produced by the upward motion of air. The lifting effect as shown in the previous section is a function of size, shape, weight, and perhaps degree of surface roughness.

The prime unit of this separator is a triangular-shaped perforated table. The table is so baffled underneath that air which is fed up through it is evenly distributed. The volume of air, which is supplied by a fan, is controllable within a wide range. A plan and front elevation of the table are shown in Fig. 7.8. The table has a reciprocating motion that moves any material upon it in the direction of conveyance. The table has vertical adjustments such that it can be tipped toward the front and toward the left, the net pitch being such that a sphere placed on the table would roll in the direction indicated by the angle-of-gravity-flow arrow. The pitch angles are shown in the discharge-edge elevation.

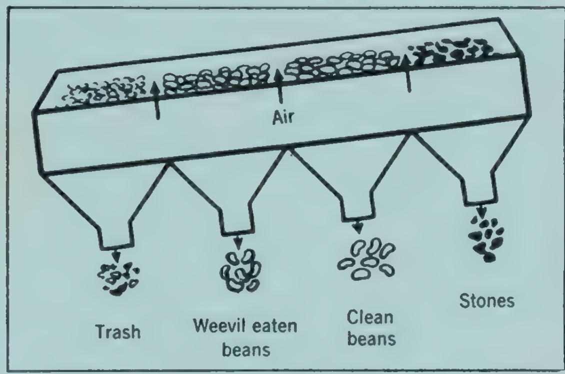
In operation, material is introduced into the feed box. Air is blown up through the perforated floor at such a rate that the material is partially lifted from contact with the floor. Lighter, smaller pieces are lifted somewhat higher and "float" down the table toward the discharge edge. The large and heavy particles are not lifted by the air. The oscillating motion of the table moves them in the direction of conveyance, and they are discharged at the right edge of the table. Other material that is

only partially lifted touches the table at frequent or infrequent intervals and is discharged at an appropriate intermediate point.

Hardware cloth, perforated corrugated metal, canvas, and other floor coverings are available to adapt the machine to various of



Top view of separating unit showing flow of material relative to angle of gravity flow and direction of conveyor.



Side view of separating unit showing materials separation, angle of incoming air, and tilt of conveyor.

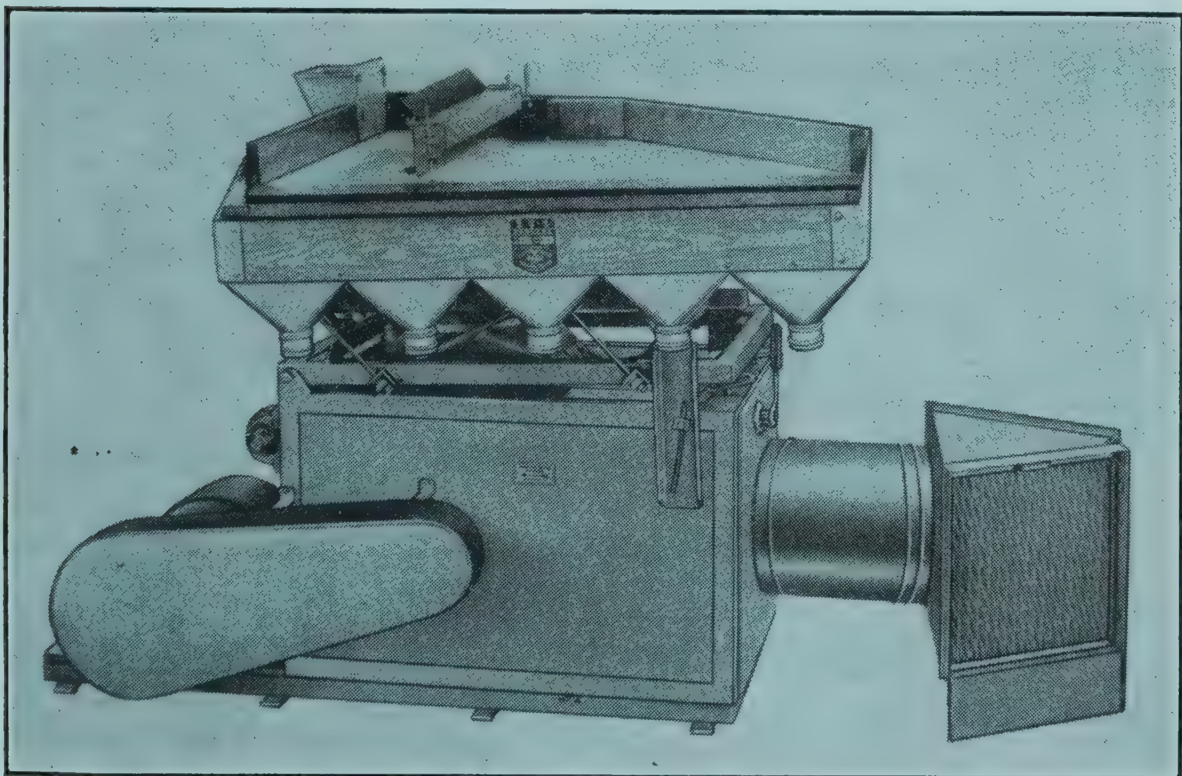


Fig. 7.8. Specific gravity separator. (*Courtesy* Sutton, Steele, and Steele Co.)

a multitude of jobs. The discharge spouts can be varied as to number and the discharge-edge distance that each serves.

For rough work where the materials to be sorted are of diverse size and weight the table pitch can be relatively steep. This

Table 7.3 REPRESENTATIVE SPECIFIC GRAVITIES FOR SOME GRAINS *

Grain	*Specific Gravity*
Wheat	1.30
Oats	0.99
Barley (white hulless)	1.33
Barley (Coast, 6 row)	1.13
Soy beans	1.18
Grain sorghum	1.24
Rye	1.23
Rice	1.12
Corn	1.19
Buckwheat	1.10
Millet	1.11

* Zink, Frank J. Specific Gravity and Air Space of Grains and Seeds. *Agricultural Engineering*, 16:439–440. 1935.

permits high capacity. If the characteristic range is small, the pitch must be less and the capacity is reduced proportionally.

With careful adjustments it is possible to separate seed from grains which cannot be separated by other devices. It is especially useful for removing lightweight infertile seed from seed stock. Germinations can be raised significantly by the gravity table.

The specific gravity separator is probably unequaled in performance, but it is expensive and its capacity (per dollar invested) is lower than that of other types of cleaners that are nearly similar in performance.

7.16. Spiral Separator. The spiral separator, Fig. 7.9, separates material on the basis of shape. The unseparated material is divided and is introduced into the inner helices at the top. The round elements in the mixture pick up speed as they roll down the helices until their centrifugal force is sufficient to cause them to roll up and over the edge. They are caught in the outer helix and roll to the bottom and out the outside spout. The elements that are not round do not roll fast enough to be discharged over the edge. They are discharged through spouts connected to the inner spirals.

Mustard, rape, vetch, wild peas, and similar round seed can be separated from wheat, flax, clover, etc. Although this device is not as versatile as the mechanical cleaners, it is simple and inexpensive and is quite useful in the seed-cleaning establishment.

7.17. Disc and Cylinder Separators. The cylinder sorter consists of a horizontal cylinder with indents on the inside surface. The indents, which are approximately hemispherical in shape, pick up grains from the mixture in the cylinder as shown in Fig. 7.10. The grains that are wider roll out of the indents before they have been lifted sufficiently to fall past the separating edge *S*. The grains of smaller width are elevated a greater distance prior to falling and are deposited in the center trough for removal or additional treatment. Separation is made on the basis of length of grain. Length is also a separating factor since long grains or foreign material in the form of sticks and stalks are not picked up by the indents.

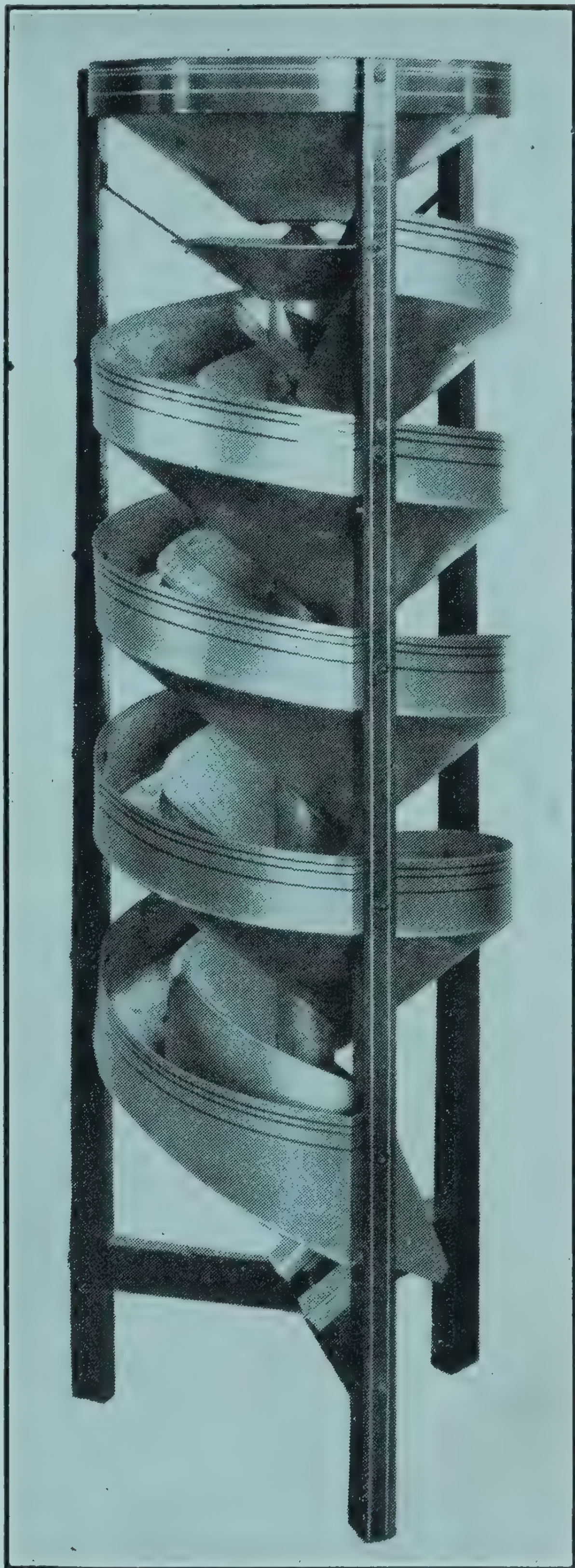


Fig. 7.9. A spiral separator used for separating dissimilarly shaped materials. (*Courtesy* Cleland Manufacturing Co.)

Fineness of separation is controlled by moving the separating edge *S*. The higher the edge, the shorter the length of the grain that is removed. The speed of the cylinder which is usually standardized by design is an important performance factor since centrifugal force, which is related to speed, tends to keep the grains in the pockets. The point of discharge is raised as the cylinder speed increases.

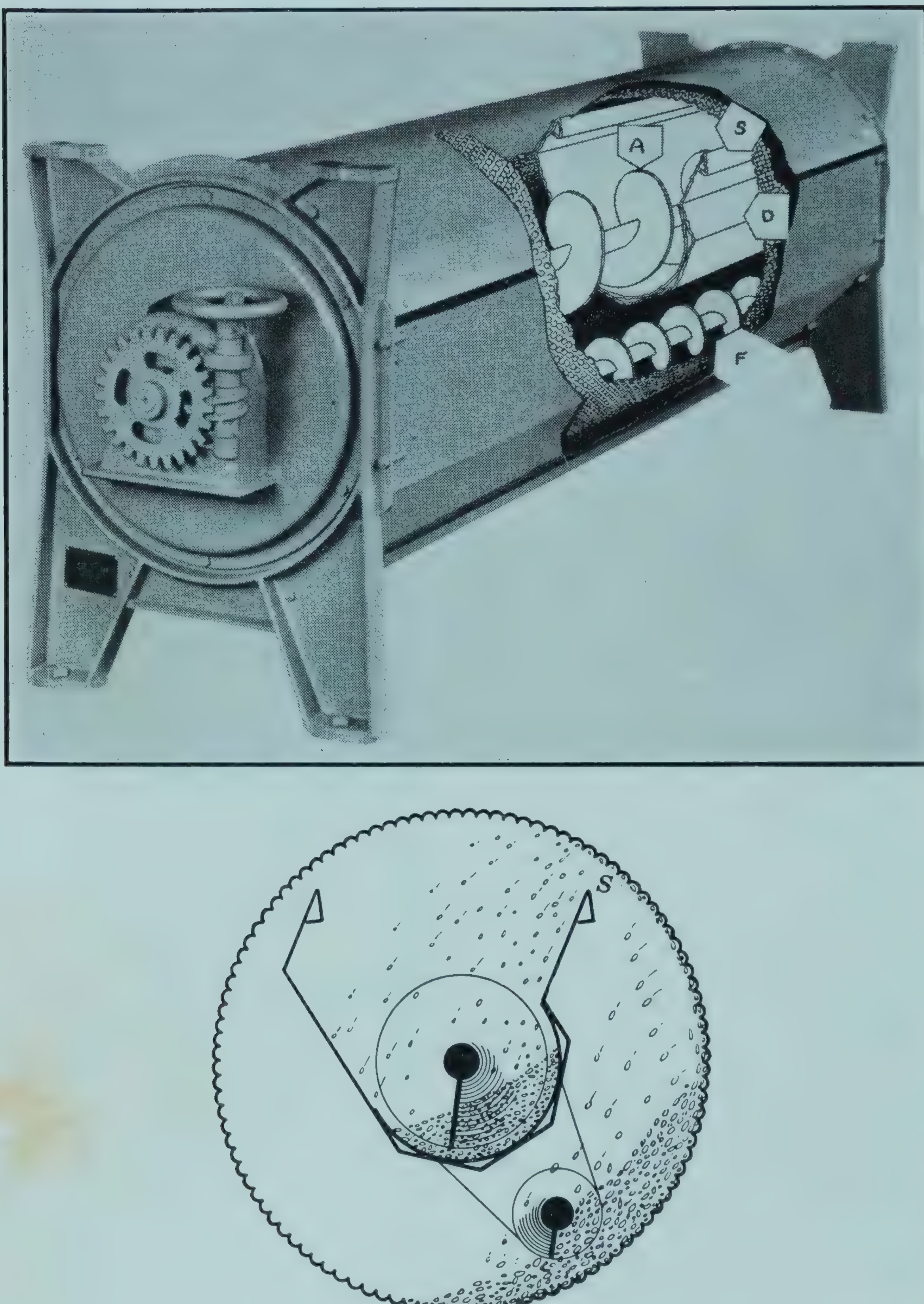


Fig. 7.10. Cross section and phantom view of a cylinder sorter. (*Courtesy* Hart-Carter Co.)

Fig. 7.11. Phantom view of a disc separator and cross section of disc pockets or indents. (*Courtesy* Hart-Carter Co.)

This sorting machine is especially useful for cleaning and sorting the grain into grade fractions, particularly the sorting, where large quantities of specific materials are handled. Milling and seed enterprises and terminal grain elevators are examples. Even though each machine is fixed as regards size of indents and speed of operation, wide flexibility of use is possible. For example, a single machine will handle cleaning and grading operations pertaining to barley, wheat, rye, and oats.

The disc separator of Fig. 7.11 separates on the basis of grain length. The pockets, which are slightly undercut as shown in the figure, can pick up and retain short grains, but long grains fall out. It is especially adapted for removing dissimilar materials. For example, wheat, rye, cockle, wild peas, mustard, wild buckwheat, pigeon grass, pin oats, and barley can be removed from oats. Similar separations can be made from other grains.

The mix to be separated is moved through the machine by flights on the disc spokes. The material (which may be either desirable or undesirable) not lifted by the discs is tailed from the end of the separator. A number of distinct separations can be made in a single machine by installing banks of discs with different characteristics. The pockets in the first bank of discs are smaller than the second bank so that the smallest material is removed first. The second bank has larger pockets than the first, and the next larger fraction is removed next. The largest grains pass through the center of the discs and are tailed from the machine.

Disc and cylinder separators have high capacity. Since all the moving parts are rotative rather than reciprocating, long life and moderate power requirements are characteristic. Each machine is fitted with a cylinder or discs having fixed characteristics. Consequently, these separators are not as versatile as certain other sorters. Even so, a single machine can be used for a sufficient number of separations to make it a general utility machine for a milling or seed-processing enterprise.

7.18. Separation Based upon Surface Texture. Surface texture may be used as a basis for separation when other methods fail.

A principle shown in Fig. 7.12 is used to separate certain rough weed seeds from useful seed of similar size, shape, and density.

The drum (1) has a special rough surface that picks up the rough-surfaced weed seeds. These seeds are thrown against the roll shield (3) and are ejected by being bounced out as indicated by the dotted line (6). Adjustments are made in rate of feed, speed of roll, character of cylinder roughness, and roll inclination.

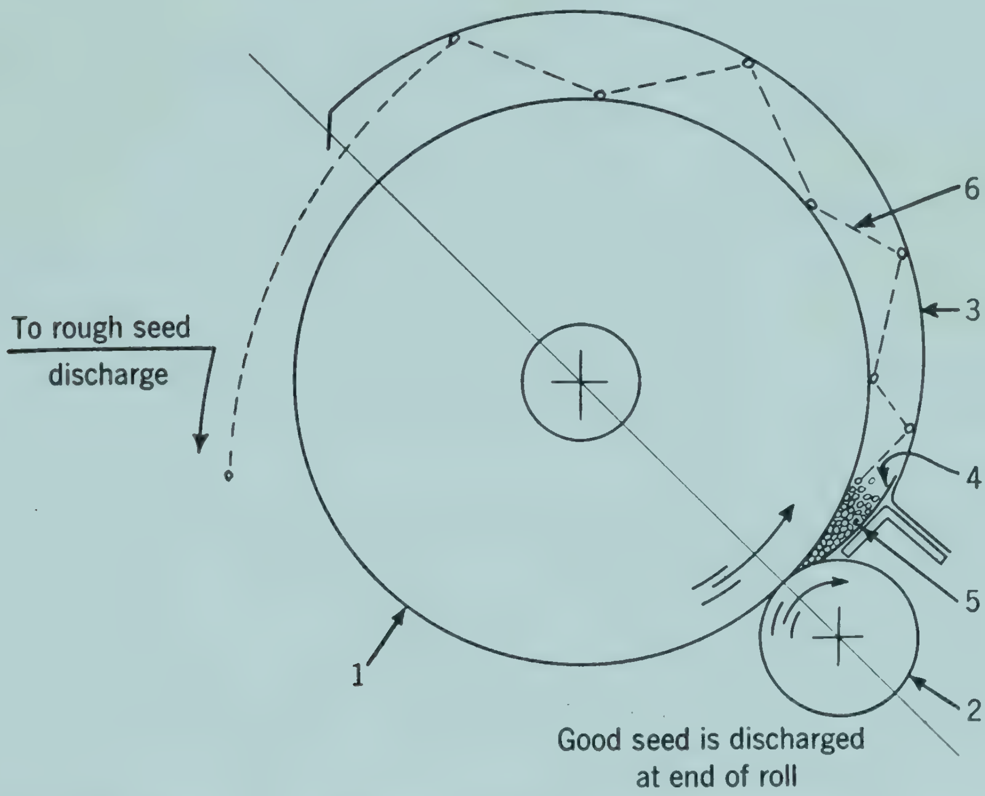


Fig. 7.12. Principle of operation of one sorter operating on the basis of surface roughness. (*Courtesy* A. T. Ferrell and Co.)

A machine developed by the California Agricultural Experiment Station to sort clods from beans operates on a comparable principle.

Some seeds develop a gummy or sticky surface when moistened with water. Others do not. Buckhorn can be separated from clover and alfalfa by moistening the mass and mixing in a small amount of finely ground sawdust. The sawdust adheres to the weed seed but not to the clover. Separation is then made by a gravity table.

7.19. Separation of Heavy and Other Foreign Substances. Stones, dirt, clay, glass, pieces of metal, etc., must be completely removed from grain or other products if the end product is to be of highest quality. This is especially important if a milling

process is to be used. Stones and metallic substances would be destructive to milling machinery and would render the end product unfit for human or, perhaps, animal consumption.

Most of the heavy foreign materials will be removed during the normal cleaning and sorting procedures, using one or more of the devices discussed previously. However, certain special procedures should be commented upon briefly.

A special gravity table combined with air flotation operating on the same principle as the gravity table discussed in sect. 7.15 can separate all heavy foreign materials. It has high capacity and is especially useful for milling houses.

Iron and steel pieces can be separated with magnetic separators.

Infrequently, hard clods are found in a product which are the same size, shape, and density as the grains of the product. Consequently, they cannot be removed by any of the conventional means.

Separation can usually be made by running the mixture between two soft rubber rolls so spaced that the material is not harmed but the clay pieces are sufficiently reduced in size to be removed by screening, aspiration, or by some other method.

If the clay pieces are too hard to be broken up by this method, the special machines discussed in sect. 7.18 are usually satisfactory.

CENTRIFUGAL SEPARATION

The familiar farm cream separator separates the cream from the skim milk by the centrifugal force resulting from rotation of the bowl. Industry uses this principle in many operations to separate a suspended material, finely divided solid or liquid, from a liquid-carrying medium. Although the cream separator is the only familiar agricultural use for this type of separation, the principles should be clearly understood so that application can be made to any possible problem.

7.20. Stokes' Equation. Stokes' equation or law defines the terminal or steady-state velocity of particles moving under an accelerational force with streamlined flow. Stokes' equation is

$$V = \frac{2r^2(\gamma_p - \gamma)a}{9\mu} \tag{7.8}$$

where V = velocity, ft per sec.
r = radius of particle, ft.
γ_p = density of particle, lb per cu ft.
γ = density of fluid medium, lb per cu ft.
a = accelerational force, lb per sec^2.*
μ = viscosity, lb per ft-sec.

If the particle density is greater than the density of the carrying medium, the particle will fall. If less than the medium, the particle will rise.

The maximum or limiting radius for which Stokes' equation applies is

$$r = \frac{1}{2} \sqrt[3]{\frac{36\mu^2}{a\gamma(\gamma_p - \gamma)}} \tag{7.9}$$

As the particle size becomes smaller and approaches the size of the molecules of the fluid medium, additional factors become effective and Stokes' law does not apply. Homogenization or emulsification provides a physical bond that makes it difficult or impossible to separate the suspended particles from the carrying medium.

In the case of butter fat suspended in skim milk, the individual particles of fat combine into clusters that have a larger effective radius than the individual particles and, consequently, rise at a faster rate. This phenomenon is characteristic of many emulsions and suspensions, oils in water and gases in liquids, for example.

7.21. The Centrifuge. Movement of dissimilar particles through a fluid can be greatly speeded by increasing the acceleration factor a, by rotating the mixture or suspension about a fixed axis. This fact is utilized in the centrifuge, which rotates a small sample of the material around an axis at a high rate of speed. Equipment of this type is familiar to laboratory technicians. The centrifuge used in determining the butter-fat content of whole milk or cream is a familiar piece of equipment in rural "cream stations."

The value of a in Stokes' equation when the settling force is due to centrifugal action is

* For free settling under gravitational force, a is g or 32.2 ft per sec^2.

$$a = \frac{(2\pi n)^2 R}{3600} \tag{7.10}$$

where n = rpm of unit.

R = radius of rotation or distance of particle from axis, ft.

By combining equations 7.8 and 7.10 we have

$$V = \frac{2r^2(\gamma_p - \gamma)}{9\mu} \times \frac{(2\pi n)^2 R}{3600} \tag{7.11}$$

or

$$V = \frac{r^2 n^2 R(\gamma_p - \gamma)}{410\mu} \tag{7.12}$$

which gives the rate of streamlined movement of a particle through a medium under the influence of centrifugal action.

7.22. The Cream Separator. The bowl, which rotates at approximately 8000 rpm, whirls the incoming milk thus producing an accelerating force that acts radially. The heavier material or skim milk will become concentrated at the outer part of the bowl, the lighter fraction or butter fat will move toward the axis of the bowl. The incoming milk causes the material to rise. The skim milk rises on the outside of the bowl and is ejected through an opening. The cone-shaped discs funnel the cream toward the center of the bowl; the cream rises and is discharged through a cream screw. The cream screw is used to regulate the rate of flow of cream which in turn regulates the concentration of butter fat; the faster the rate of flow, the smaller the concentration of butter fat.

A particle or globule of butter fat is from 4 to 8 microns * in diameter and weighs 54 lb per cu ft. Skim milk weighs 64.4 lb per cu ft.

Using equation 7.8 the rate of rise of the fat (theoretical) under the action of gravity would be, if the diameter were 6 microns,

$$V = \frac{2(0.0000099)^2(64.4 - 54)32.2}{9 \times 0.000922}$$

$$= 0.0000079 \text{ ft per sec or } 0.028 \text{ ft per hr}$$

On the basis of this figure, 18 hr would be necessary for cream to rise 6 in. in a milk bottle. Although this is representative of

* A micron is 0.001 of a millimeter or 0.00003937 in.

observed laboratory rates, actual rates are much higher than this, owing, assumably, to clumping of the particles.

Now consider the action in a cream separator having a bowl 6 in. in diameter and operating at 8600 rpm. Using the combined equation 7.12 and assuming that the average reaction is 2 in. from the bowl axis, we find

$$V = \frac{(0.0000099)^2(8600)^2 0.16(64.6 - 54)}{410 \times 0.000922}$$

$$= 0.032 \text{ ft per sec or } 0.38 \text{ in. per sec}$$

The actual rate would be higher than this due to clumping.

CYCLONE SEPARATOR

The cyclone separator or collector is used extensively in processing and other operations as a device for collecting the end product. It is also frequently used in connection with pneumatic conveying of products and wastes from processing.

7.23. Theory. The basis of operation can be shown from Fig. 7.13. The air and material enter tangentially at the top and descend with a circular motion described by an outer vortex. The material is separated during the downward descent, and the clean air ascends in a tighter vortex at the center and is discharged.

A particle that has entered the cyclone is acted upon by two forces, C_f and W. The centrifugal force, C_f, which acts upon the particle is

$$C_f = WV^2/gR \tag{7.13}$$

where W = weight of particle, lb.
V = linear or tangential velocity, ft per sec.
g = acceleration of gravity, 32.2 ft per sec^2.
R = radius of rotation, ft.

Therefore, the separating force F is

$$F = W\sqrt{V^2/gR} + 1 \tag{7.14}$$

The performance factor S is

$$S = C_f/W = V^2/gR \tag{7.15}$$

The larger S is, the more effective separation will be. S is an acceleration multiplier and defines the number of "g's" acting on a particle. Note that this factor increases directly as the square of the velocity and inversely as the radius of rotation.

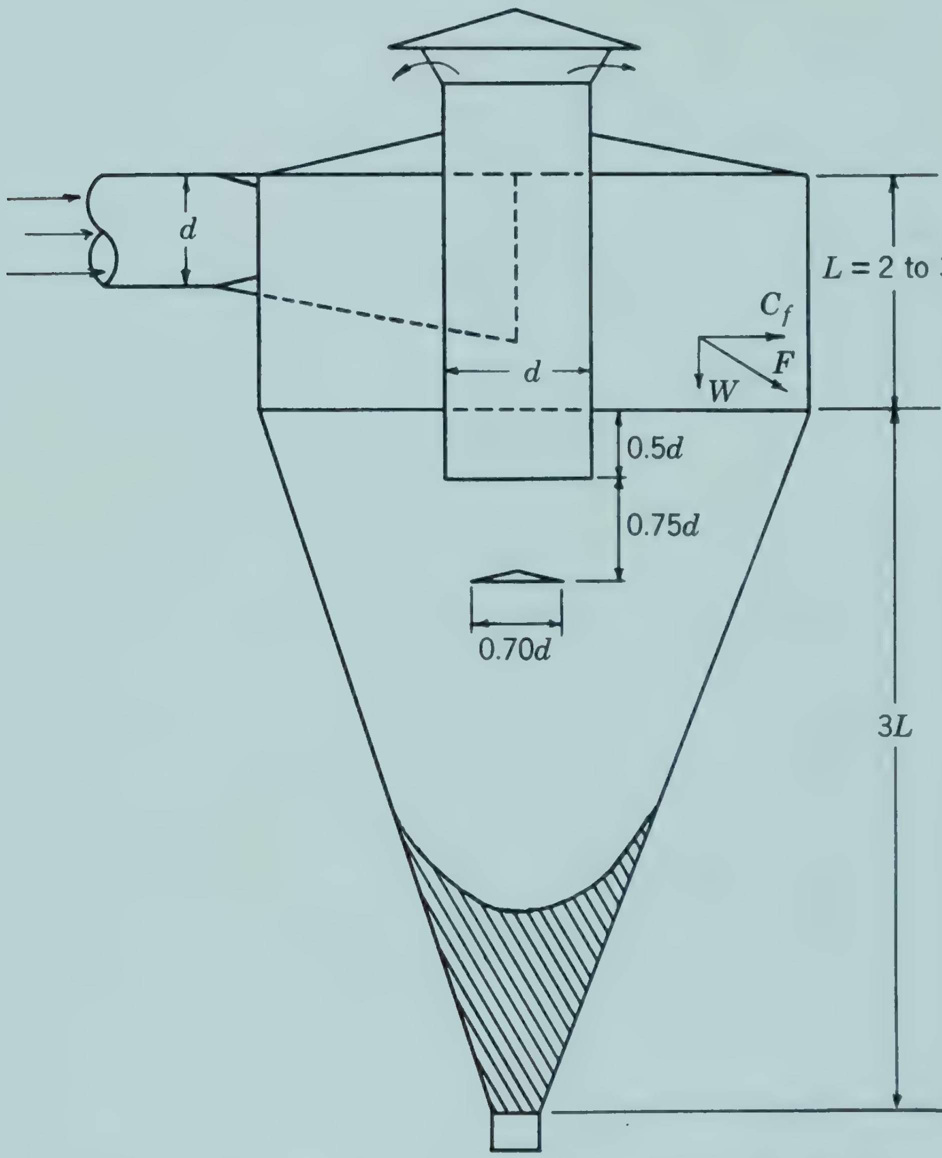


Fig. 7.13. Schematic drawing of a cyclone separator showing relative recommended dimensions. Note that the over-all diameter is independent of the proportions shown. The over-all diameter will depend upon the quantity of material to be handled and smallest sized particle to be removed.

The entering particle is acted upon by the force F which causes it to move outward toward the wall during its downward helical travel. As it approaches the wall, the velocity V decreases because of wall friction and the particle settles into the cone. Its rate of movement is a function of the separation factor S and the weight and size of the particle. The terminal velocity or settling rate of finely divided materials depends upon weight and effective

size. The lighter and smaller they are, the longer it takes for them to settle a specified distance. Consequently, the depth or number of turns in the helix is also important.

The actual number of effective turns in a specific separator is usually difficult to estimate. A safe assumption for a common cyclone is two, but the performing range in many instances may be less.

If a symmetrical flow pattern is assumed, the effective circumferential velocity of the outer and inner or descending and ascending helices can be shown related to the outer o and inner i cylinder diameters by equation 7.16.

$$V_i/V_o = (d_o^2 - d_i^2)/d_i^2 \tag{7.16}$$

The ratio of the performance factors can then be shown related to the diameters thus:

$$S_i/S_o = d_o(d_o^2 - d_i^2)^2/d^5$$

7.24. Design. Unfortunately, even though the theory of the cyclonic separator is known, few data are available which will aid in the critical design of an efficient unit for a specific job.

The procedures of sect. 7.12 can be used to assist in designing a cyclone for a specific job. The radial velocity of the particle is V for terminal conditions, and the gravity factor g is multiplied by the separating coefficient S, equation 7.15. The time required to accelerate to this velocity may be considered negligible. The separating distance is the width of the inlet, and the time available for separation is the time required for an element of air to move through the effective outer helix. The particle may be considered as moving in a true radial direction and at the same circumferential velocity as the carrying air.

Two general types of cyclone separators are recognized. The common large cyclone, Fig. 7.13, is satisfactory for materials larger than 300 mesh. This would include nearly all the materials that the agricultural engineer would handle. Deep narrow separators usually installed in banks attached in parallel will separate such finely divided substances as flour, powdered milk, fly ash, etc.

General relative dimensions for a large-sized cone by Dalla Valle [4] are shown in Fig. 7.13. The inverted cone placed below

the inner cylinder has been found to improve efficiency, but no theoretical basis for the improvement is known.

The inlet should be gradually changed from circular to rectangular, the rectangular opening having the same area as the round

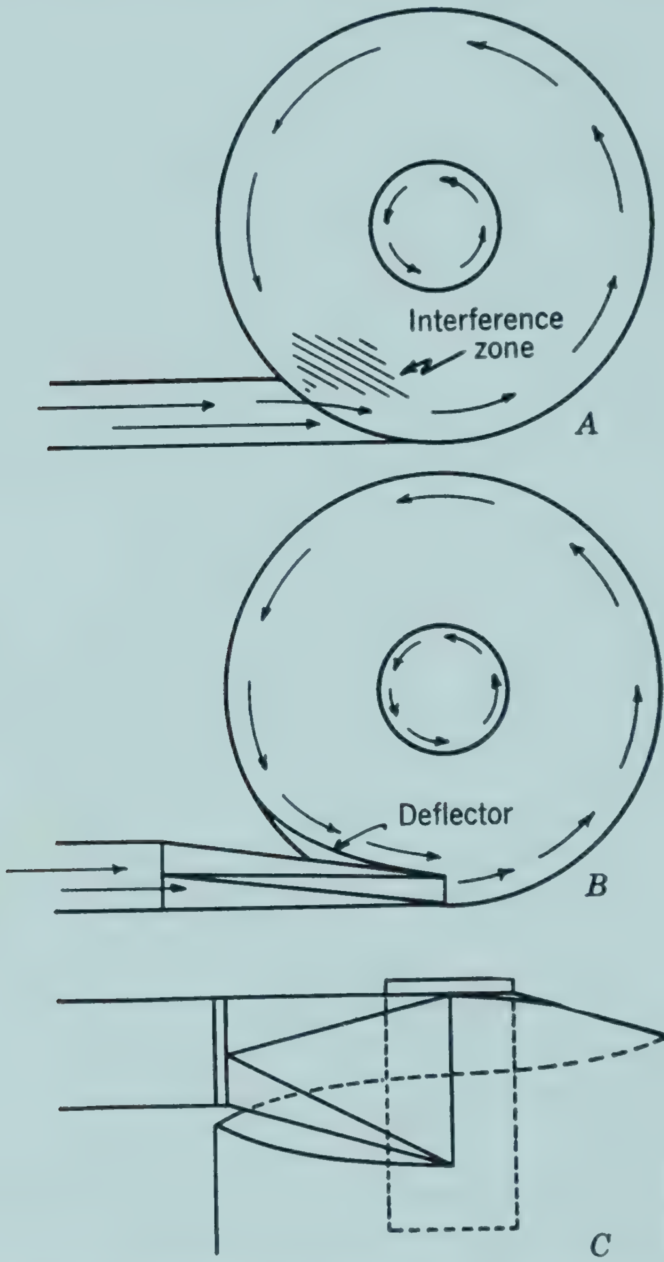


Fig. 7.14. The pressure drop through a cyclone separator can be reduced by using a deflector or a helical top to minimize the interference zone.

approach pipe. The outer portion of the pipe and reducer should be tangent to the cylinder as shown in Fig. 7.14*b*, and the width of the inlet should be as small as practicable.

Although pressure drop through a cyclone can be calculated on the basis of inlet friction and outlet losses, practice has shown there is little correlation between calculated and actual losses. These figures vary from 0.5 to 5 or more pressure heads with

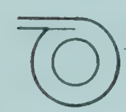
errors as high as 500 per cent. Interference between various spirals or turns in the air stream within the cyclone is believed responsible for the deviation. Interference at the point of air entry is particularly critical. This loss can be reduced by installing a deflector, Fig. 7.14*b* and/or providing a helical-shaped top as shown in Fig. 7.14*c*. These details are not recommended unless pressure drop is an important factor. If the deflector is not needed the inlet should be flush with the periphery, that is, terminated at the wall as in Fig. 7.14*a*. Back pressures in helical-top cyclones may be less than 0.6 the inlet velocity head.

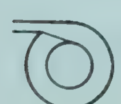
Melvin W. First developed an equation for pressure loss through a cyclone which appears to be acceptable for many cyclones (*Fundamental Factors in the Design of Cyclone Dust Collectors,* an unpublished thesis, Harvard University, 1950). This equation is:

$$P = \frac{12bh}{KE^2(L/d)^{1/3}(H/d)^{1/3}}$$

where P = pressure drop, number of inlet velocity heads.
d = cylinder diameter, ft.
L = cylinder height, ft.
H = cone height, ft.
b = entry width, ft.
h = entry height, ft.
E = exit-duct diameter, ft.
K = vane constant, dimensionless.


K = 0.5 for no inlet vane.


K = 1.0 for inlet vane that does not expand entering air streams or touch exit duct.


K = 2.0 for entry vanes that expand entering stream and extend from entry to exit-duct wall.

Other than these general comments and precautions, no specific procedures can be suggested to insure low pressure losses.

The angle of the main spiral is usually between 10° and 15°. The inlet velocity should be low to minimize pressure loss and high for effective separation. Velocities from 20 to 70 ft per sec are usual with 50 about optimum.

Dalla Valle [4] reports work by Rosin, Rammler, and Intelmann, which gives the theoretical equation for the smallest particle removed by a cyclone.

$$d' = \left(\frac{9\mu d}{2\pi NV(\gamma_p - \gamma)(4R/d)^k}\right)^{1/2} \tag{7.17}$$

where d' = particle diameter, in.
μ = viscosity of air, lb per ft-sec.
d = diameter of exit duct.
N = number of effective turns of air stream in the inner spiral.
V = entrance velocity.
γ_p = density of particle.
γ = density of air.
R = radius at which the spiral velocity is equal to v.
k = constant, 0.5–0.7.

This equation presumes streamline flow.

REFERENCES

1. Alden, J. L. Industrial Exhaust Ventilation. Cyclones and Filters. *Heating and Vent.* 35:48–53. July, 1938.
2. Banton, R. *Problems in Cleaning Seed. Some Factors in Effective Operation of Seed Cleaners Using Air and Screens, with Suggestions on Correct Use of Screens for Many Separations.* A. T. Ferrell & Co., Saginaw, Michigan. 1946.
3. Cruess, W. F. *Commercial Fruit and Vegetable Products.* Second Edition. McGraw-Hill. 1938.
4. Dalla Valle, J. M. *Micromeritics. The Technology of Fine Particles.* Pitman. Second Edition. 1948.
5. Lapple, C. E., and Shepherd, C. B. Calculation of Particle Trajectories. *Ind. Eng. Chem.* 32:605–617. 1940.
6. Perry, R. L., et al. Fruit Dehydration. Principles and Equipment. *Calif. Agr. Expt. Sta. Bull.* 698. 1946.
7. Roadhouse, C. L., and Henderson, J. L. The Market-Milk Industry. McGraw-Hill. 1941.
8. Shepherd, C. B., and Lapple, C. E. Flow Pattern and Pressure Drop in Cyclone Dust Collectors. *Ind. Eng. Chem.* 31:972–984. 1939.
9. Vegetable and Fruit Dehydration. A Manual for Plant Operators. *U.S.D.A. Misc. Publ.* 540. 1944.

PROBLEMS

1. Analyze or discuss the motion of a sieve, Fig. 7.4, with ⅛-in. holes on 3/16-in. centers, diamond arrangement, from the standpoint of:

a. Proper V and H for the oscillating motion.
b. The optimum screen pitch.
c. Statistical probability of a seed passing a hole during a 2-ft downward travel.
d. Effect of seed-layer depth on performance.

2. A ladino clover sample weighs 0.100 g per 160 seeds. Assume each seed is equivalent to a sphere 0.95 mm in diameter. Estimate the flotation air velocity.
3. A cylinder separator is 18 in. in diameter.

a. At what rpm would all the material be held in the indents by centrifugal force?
b. At what rpm would it be discharged at 120° from the bottom?

4. A cyclone 6 ft in diameter with an inlet 1 ft in diameter is designed as shown in Fig. 7.13 with $L = 2.5d$. The inlet is 4 in. wide, and inlet velocity is 50 ft per sec.

a. Determine the smallest particle which can be collected.
b. Estimate the pressure drop through the unit.

5. Determine the diameter of a cyclone proportioned as in the text for collecting alfalfa meal. Conditions:

Inlet width	¼ cyclone diameter
Entrance air velocity	50 ft per sec
Smallest particle	0.0012 in. in diameter
Particle S.G.	1.10
Separating height	1¼ diameters
Helix pitch	15°

Procedure: Determine the required radial particle velocity. Calculate the Reynolds number and find CRe^2. Then combine equations 7.7 and 7.15 and solve for the radius R which is half the required diameter.

CHAPTER 8

Materials Handling

NOMENCLATURE

A = an empirical constant, dimensionless.
a = pressure drop, system handling air only, in. water.
B = an empirical constant, dimensionless.
C = conveyor capacity, cu ft per min.
D = screw diameter, in.
d = shaft diameter, in.
F = a material factor, dimensionless.
F_c = coefficient of friction, flights and chain.
F_m = coefficient of friction, material.
g = acceleration of gravity, 32.2 ft per sec per sec.
H = lift, ft.
h = a vertical distance, ft.
K = a variable, related to velocity.
L = horizontal projected length of loaded conveyor, ft.
L_c = horizontal length, ft.
M = margin, in.
M_c = a proportionality constant.
m = pressure drop, in. water.
N = revolutions per minute.
P = screw pitch, in.
p = velocity pressure, in. water.
Q = material rate, lb per min.
R = a ratio, lb material per lb air.
r = radius, ft.
S = centrifugal force, lb.
t = time, sec.
V = velocity, ft per min.
v = speed, ft per min.
W = unit volume weight, lb.
W_c = weight of flights and chain, lb per ft.
w = flight width, in.

The performance of a processing plant is measurably affected by the efficiency of the movement of materials from one unit operation to another. The importance of this movement of ma-

terials is not essentially a function of its magnitude. Efficiency in delivery of grain to a hammer mill and removal of the ground product in a grinding job on the farm are just as important, relatively, as movement of material through a packing plant.

Industrial materials handling is a highly specialized enterprise. Its procedures and details have developed out of usage and experience in great measure, the rational approach being undeveloped in many cases. Large installations should be designed and installed by materials-handling engineers. Smaller, less involved installations can be designed and installed by the resident engineer of a processing plant, a local mechanic, or the farmer.

Materials handling as recognized in general implies the movement of materials in any direction and, consequently, includes elevation as well as movement in a horizontal plane. Movement of fluids could be considered under this heading, but since they have been discussed in the chapters dealing with fluid mechanics, flow-rate measurement, pumps, and fans, fluids will be omitted.

Handling devices may be classified as follows:

1. Belt conveyors.
2. Chain conveyors.
3. Screw conveyors.
4. Bucket elevators.
5. Pneumatic conveyors.
6. Gravity conveyors.
7. Cranes.
8. Lift and carrying trucks and carts.

Cranes, trucks, and carts might be considered as intermittent conveyors. These devices are frequently useful in providing an efficient flow of material through a plant and, consequently, are discussed in this connection.

BELT CONVEYORS

The belt conveyor is essentially an endless belt operating between two or more pulleys. The belt and its load are usually supported on idlers. The installation may be a simple one such as a light canvas belt sliding over a long table and carries fruit or a very heavy belt that is supported by antifriction bearings and carries grain.

8.1. Characteristics. Belt conveyors have a high mechanical efficiency since, in larger installations, all the load is carried on antifriction bearings. Damage to the product being transported is slight since there is little or no relative motion between the product being carried and the belt. The carrying capacity is high since relatively high belt speeds are possible. Materials can be conveyed long distances, but there is a limit to the angle of elevation. A properly designed and maintained belt system has a long service life, but the initial cost is usually high. Installation is advisable only when amortization of the high initial cost can be assured.

8.2. Details. The elements that must be considered relative to belt conveyors are the belt, drive, tension or take-up feature, idlers, and loading and discharge devices.

Belts must be flexible enough to conform to the pulleys, wide enough to carry the quantity and type of material required, have strength enough to stand up under the expected load and operating tension, and a resistant surface. Stitched canvas, solid-woven, balata, and rubber belts may be used. Stitched canvas and woven belts are usually impregnated with a waterproofing material. A rubber belt is made of canvas or woven material impregnated and vulcanized with rubber and covered with a rubber sheet. Balata belts are similar to rubber belts as regards aging but are affected by temperatures over 120°F.

The drive should be at the discharge end of the belt and can be a conventional belt drive. The pulley must be large enough to provide enough contact surface with the belt to insure a positive drive. Additional contact surface may be provided by using an idler pulley to provide more wrap contact of the belt. Pulley diameters must be large enough to keep from overflexing the belt.

The take-up that is necessary because of stretch of the belt and of contraction and expansion due to changes in moisture and temperature can be manual by adjusting screws or automatic by attaching a dead weight. This adjustment can be on the foot end pulley or on an idler pulley.

The load-carrying idler pulleys, Fig. 8.1, may be plain wood or light steel for flat belts or multiple when advisable to provide a trough to increase the carrying capacity or to confine materials that would not stay on a flat belt.

The material can be fed onto the belt by hand or by a mechanism that provides a continuous steady flow. This may be a simple funnel with gate valve or if the material is not free flowing, an apron, screw, vibrating, or other type of feeder may be necessary.

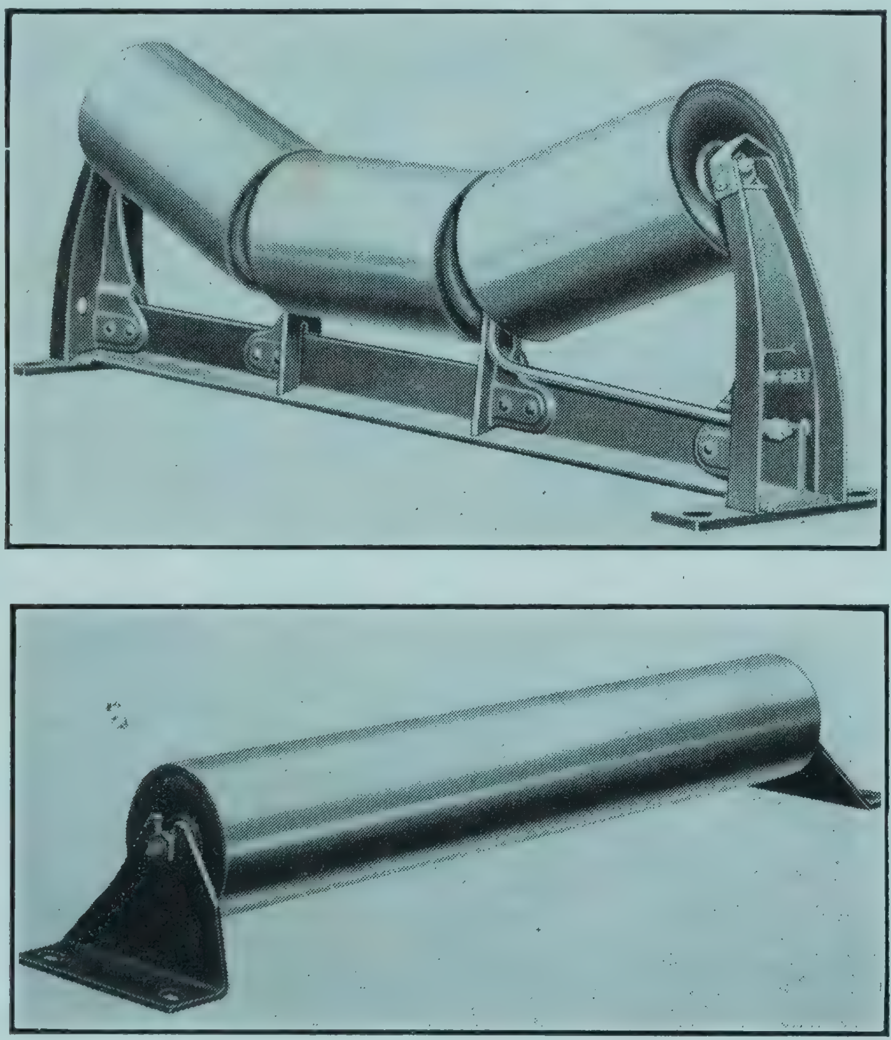


Fig. 8.1. Troughed and flat idler pulleys for belt conveyors. The straight pulley is used for the empty belt return and infrequently for carrying the load. (*Courtesy* Link-Belt Co.)

The material may be discharged over the end of the belt, by a diagonal scraper, by tilting of one or more of the idler pulleys, or by a tripper. A tripper consists essentially of two idler pulleys that cause the belt to take the shape of an S, Fig. 8.2. The material is discharged over the top pulley and is caught by a chute that diverts it to one side of the belt or the other. A short belt conveyor operating at right angles to the main belt may replace the chute if it is desirable to move the material a considerable distance from the main belt.

Trippers and scrapers are usually movable so that discharge may be made at any point on the belt or on either side. They may be reversible so the main belt can operate in either direction. If discharge is not desired the material is permitted to fall from the upper pulley back to the belt.

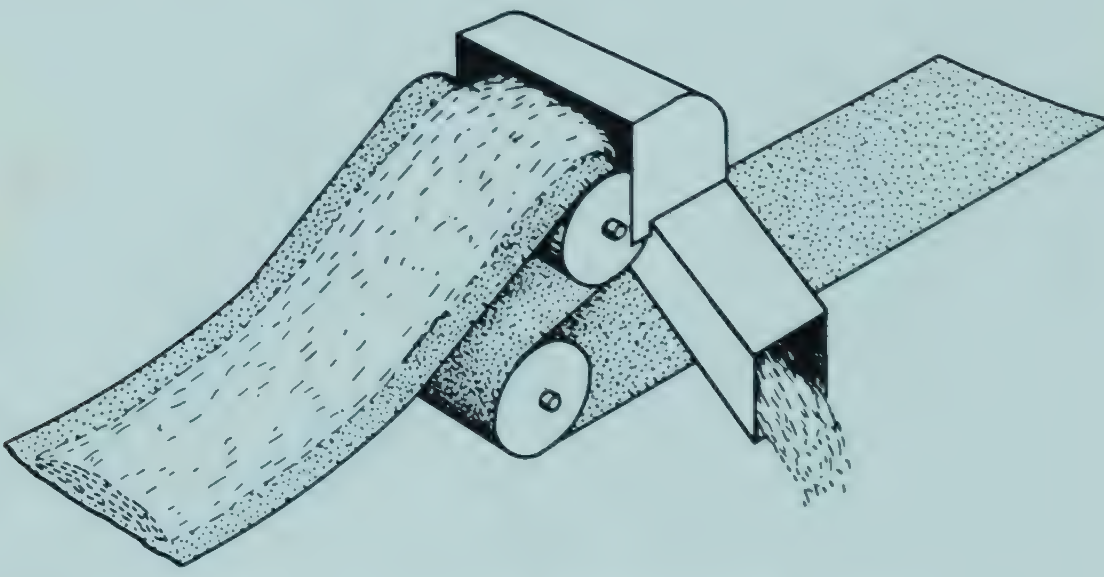


Fig. 8.2. Operational principle of the tripper.

Discharge by tipping idlers may not be advisable since the material is discharged over a considerable length of belt and there is an additional strain placed upon the belt due to twisting. An angle scraper is the simplest discharging device and is satisfactory for many materials. The various dischargers are designed so that they can be controlled remotely.

8.3. Design. The following suggestions are for preliminary calculations only. A large belt-conveyor system is an expensive

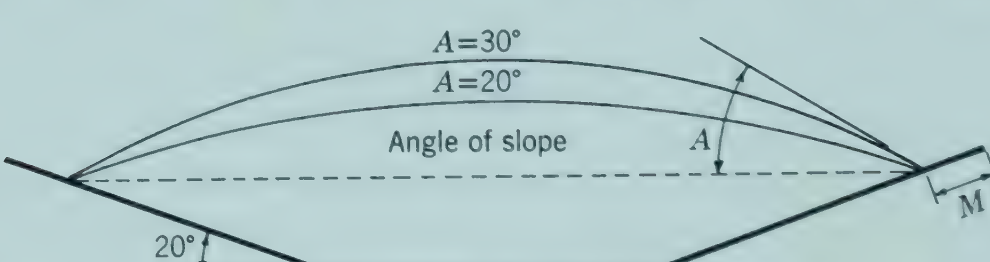


Fig. 8.3. Cross section of loaded belt showing the surcharge—material above the dashed line—and the top profile for various slope angles.

and intricate installation and should be designed and installed under the supervision of specially trained conveyor engineers.

The width of the belt is determined by the size or amount of material to be conveyed, the quantity to be conveyed, and the

type of service. The load cross section of a troughed belt is shown in Fig. 8.3; and the areas of load cross section, in Table 8.1 adapted from Hetzel and Albright.[4] The surcharge is that portion of the load above the level indicated by the dotted line in Fig. 8.3 and may be considered as the load on a comparable flat belt. The surcharge angle is large for large lumped materials, particularly if mixed with fine material, such as mine-run coal. The 20 degree arc shows the surface profile of most materials.

Example. A troughed belt with a 20 degree surcharge is to convey 1500 bu of wheat (45 ton) per hr. What width of belt should be used if maximum operating speed is assumed? The calculation is:

$$1500 = (\text{Area of cross section in sq ft} \times \text{Speed in ft per min} \times 60)/1.25$$

By trial and error, using the cross-section area from Fig. 8.3 and the maximum speed from Table 8.1 for consecutive belt widths, a 14-in. belt traveling at 400 ft per min is found to give

$$(0.096 \times 400 \times 60)/1.25 = 1830 \text{ bu per hr}$$

which is the best value to select.

Table 8.1 BELT-LOAD CROSS-SECTION AREAS AND MAXIMUM BELT SPEEDS

Belt Width, in.	*Clear Margin (M), in.*	*Total Cross-Section Area, sq ft, for Surcharge Angle A*			*Max. Speed, ft per min*	
		10°	*20°*	*30°*	*Nonabrasive Fine Materials*	*Grain*
14	1.7	0.074	0.096	0.117	300	400
16	1.8	0.101	0.131	0.162	300	450
18	1.9	0.134	0.173	0.214	400	450
20	2.0	0.170	0.220	0.272	400	500
24	2.2	0.257	0.332	0.410	500	600
30	2.5	0.421	0.542	0.669	550	700
36	2.8	0.622	0.803	0.991	600	800
42	3.1	0.869	1.12	1.37	600	800
48	3.4	1.16	1.48	1.83	600	800
54	3.7	1.45	1.90	2.33	600	800
60	4.0	1.83	2.36	2.91	600	800

Fine free-flowing materials may be blown or shaken off the conveyor at high belt speeds, particularly if a flat belt is used.

Belt incline is limited to 15°–17° for grain, 18°–20° for bank run gravel, and 20°–23° for earth and lime.

The horsepower required for movement of materials by belt conveyors can be calculated by conventional engineering methods by considering the lift, the frictional resistance of the belt, and the frictional resistance of the various pulleys and tripping devices. However, the constants used in such a procedure vary with change in operating conditions; also, flexing of the load and belt between supporting pulleys absorbs some energy. Empirical data have been found more applicable than rational data.

The Link-Belt Co.[7] has found that power for their standard installations can be calculated from the following equations:

Horsepower to drive empty conveyor

$$= \text{Belt speed, feet per minute } (A + BL)/100 \quad (8.1)$$

L is the conveyor length in feet. Constants A and B depend upon the belt width and are given in the accompanying table.

Conveyor Belt Width, in.	A	B
14	0.20	0.00140
16	0.25	0.00140
18	0.30	0.00162
20	0.30	0.00187
24	0.36	0.00224
30	0.48	0.00298
36	0.64	0.00396
42	0.72	0.00458
48	0.88	0.00538
54	1.00	0.00620
60	1.05	0.00765

Horsepower to convey material on level

$$= \text{Tons material per hour, } (0.48 + 0.00302L)/100 \quad (8.2)$$

Horsepower to lift material

$$= \text{Lift in feet} \times 1.015 \times \text{Tons of material per hour}/1000 \quad (8.3)$$

The total power required is the sum of the powers calculated from equations 8.1, 8.2, 8.3.

Example. In the previous example, the conveyor is 400 ft long and has a 15° incline. What is the horsepower requirement?

From equation 8.1 the power required to drive the empty conveyor is:

$$400(0.20 + 0.00140 \times 400)/100 = 3.04 \text{ hp}$$

From equation 8.2 horsepower required for conveying is:

$$45(0.48 + 0.00302 \times 400)/100 = 0.76$$

The lift is 400 sin 15° = 104 ft. The power required to elevate the material is determined thus from equation 8.3:

$$104 \times 1.015 \times \tfrac{45}{1000} = 4.75 \text{ hp}$$

Therefore, the total horsepower requirement is

$$3.04 + 4.75 + 0.76 = 8.55$$

This value does not include the power required for trippers and other auxiliary equipment.

CHAIN CONVEYORS

Chain conveyors used in agricultural processing may be contrasted to belt conveyors in many ways. Belt conveyors are expensive, quiet, fast, mechanically efficient, and must be carefully engineered to insure satisfactory performance. On the other hand, chain conveyors are not so expensive, may be noisy, are slow, are not mechanically efficient, and do not require as specialized skill for design. Because of versatility in design and the advantages indicated above, chain conveyors are admirably suited to a great variety of materials moving jobs. This is particularly true in agriculture where service is intermittent.

Chain conveyors may be classified in three ways: (1) trolley, also called overhead or monorail, (2) scraper, and (3) apron.

8.4. Trolley Conveyors. This type of conveyor consists of an overhead I-beam track with trolleys that are fastened together by chain, Fig. 8.4. The trolley conveyor can be used for products of large unit size or for those that are handled in boxes or baskets at some time during processing. Meat products, fruits, and vegetables are examples of materials that may be handled in this manner.

The direction of motion of a conveyor of this type is extremely flexible. It can be designed to make sharp turns up to 180°. Steep elevations can be included in the track, the incline being limited only by the clearance of the equipment and the load. This feature is particularly useful where the product must be immersed in a bath for such operations as blanching, cooking, or cooling.

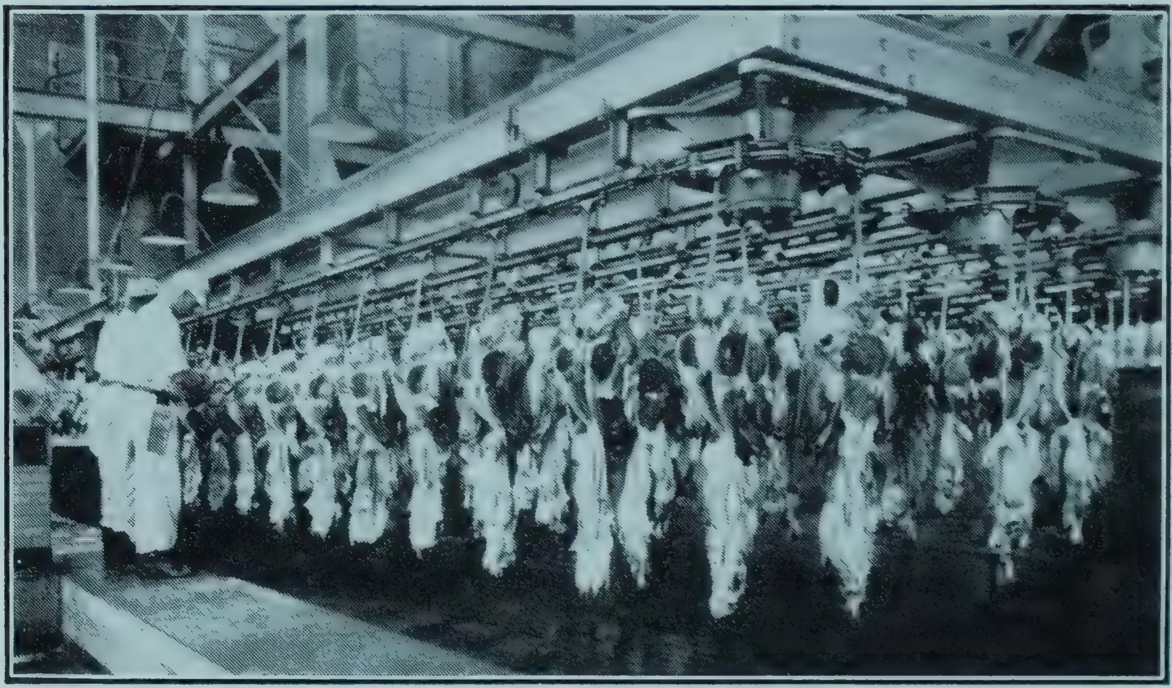


Fig. 8.4. A trolley conveyor in a meat processing plant. (*Courtesy* Link-Belt Co.)

8.5. Scraper Conveyors. Scraper conveyors used for granular, nonabrasive materials are simple, cheap, easily constructed, and may operate at steep inclines. However, power requirements are high and wear may be excessive.

Scraper conveyors are used extensively for moving raw products, beets, potatoes, small grains, for example, into processing plants. They may be permanent or portable, the portable farm grain elevator being the most widely used adaptation.

The types of chain available for conveying are extremely varied. A simple complete classification would be difficult to provide. The main types, however, are:

1. Malleable detachable.
2. Malleable pintle.
3. Steel.
4. Roller.
5. Combination.

The malleable detachable chain is the most common. It is used for light intermittent service such as portable farm grain elevators and elevators on threshing and grain-cleaning machinery. Pintle chain characterized by a pin that connects the links is used for more rigorous service such as vertical inside grain elevators, beet elevators, and materials elevators for processing plants. Steel

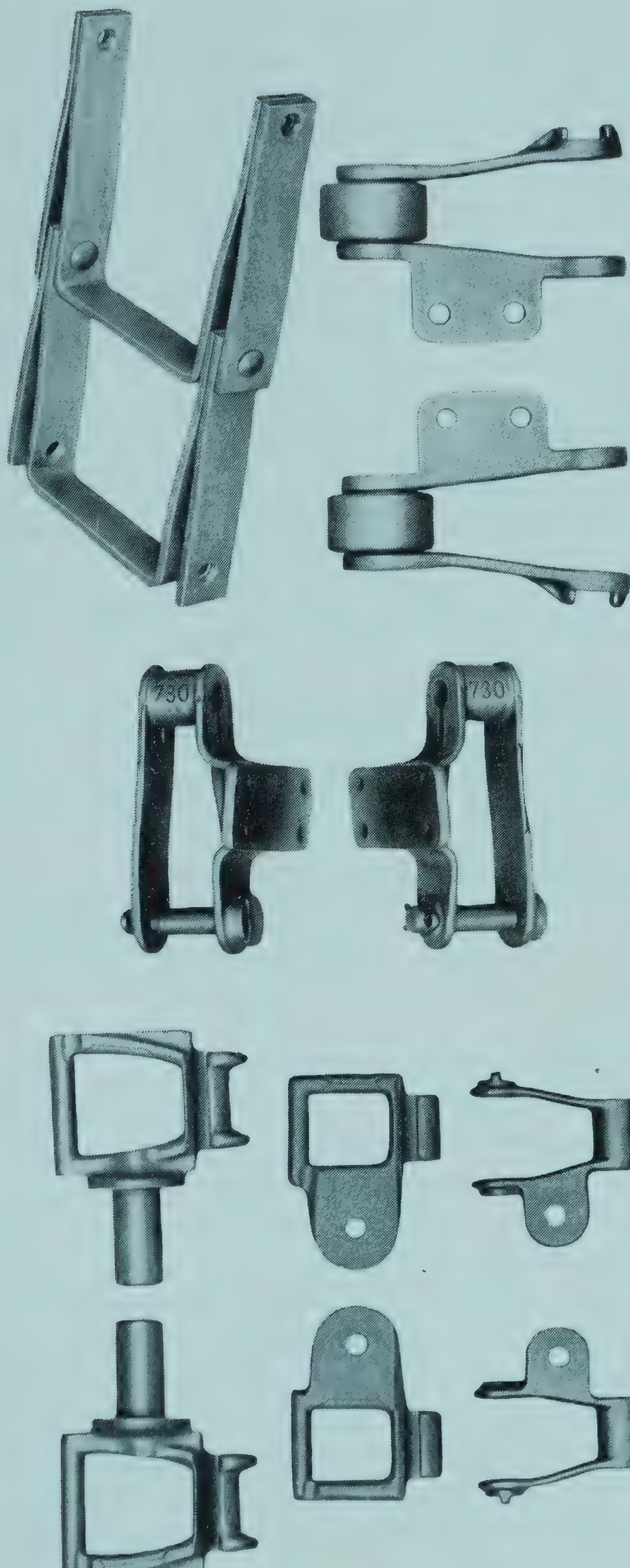


Fig. 8.5. A few types of conveyor chains. (*Courtesy* Link-Belt Co.)

chain is used where high strength or good wearing qualities, or both, are needed. Roller chain is fitted with rollers or wheels to minimize friction and reduce wear. Combination chain is made in such a way that various features of the above three types are combined to provide certain definite performance characteristics. Chains of special alloys are available for operation in the presence of heat, chemicals, abrasive substances, etc.

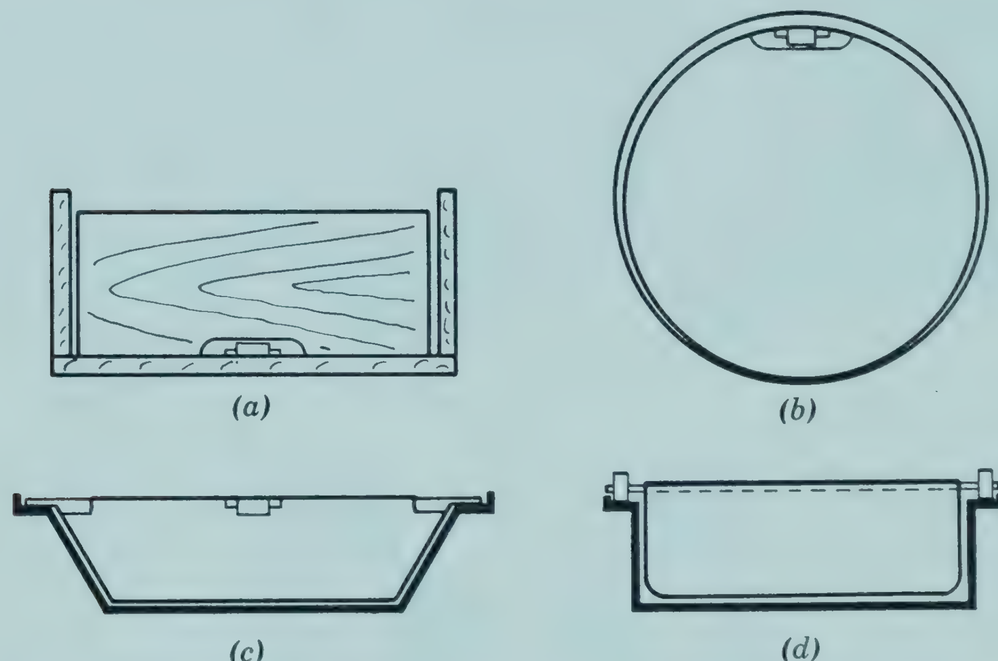


Fig. 8.6. Cross sections of some scraper conveyors. Conveyor *a* may be made of wood, as shown, or steel. If closed at the top, it may operate in the normal or inverted position. Conveyor *b* is a steel pipe with a cylindrical flight. Power and wear are minimized by supporting the flights on wearing plates *c* or on rollers *d*.

A few chain-link types are shown in Fig. 8.5. The lugs are designed to fasten to flights of various type, or, in some instances, to contact the material directly and move it without the benefit of flights, transfer chain for example. The links may be fitted with rollers that carry the load and minimize frictional resistance.

The simplest conveyor is one made with "sawdust" chain, the links acting as flights, Fig. 8.5. As the name implies, it is used particularly for removing sawdust from sawing and milling operations, but it can be used for many other materials such as hulls, husks, and pulp where the quantities involved are not high.

Scraper conveyors with attached flights are designed in a variety of ways. The simplest of them is a single chain with flights operating in a wood or steel trough. Elevators of this type are

used extensively for moving the products of farming operations. Cross sections of this and other conveyors are shown in Fig. 8.6. Materials of large granular size are conveyed on the top of the conveyor. For small granular material, the lower flights are enclosed and the material is conveyed at this point. This permits greater capacities because the trough can operate full and at a higher speed than topside movement permits. Also, the enclosed feature permits high elevation angles. Discharge for both top- and bottom-run conveying is usually at the head shaft. However, by putting gates in the lower trough, discharge for lower flight conveying can be made at any point in the conveyor. Similarly, top flight discharge can be facilitated by gates in the top trough and an open or skeleton lower trough.

8.6. Apron Conveyors. If the flights in the scraper conveyors are replaced with flat slats, steel plates, or boards, we have essentially a moving platform or apron. This type of device can be used for conveying sacked materials and materials of large unit size.

8.7. Chain Conveyor Design. The following design procedure applies to scraper conveyors, but the student will note that apron and trolley conveyors can be designed in a comparable manner.

Flight height, length, and spacing will depend upon the expected duty of the conveyor. Flat flights would be recommended for sacked material, shallow flights for large-unit-size material such as ear corn and sugar beets. For small grains and comparable materials, flight height should generally be approximately 0.4 the flight length and spaced at approximately the length.

Flight speeds vary from 75 to 125 ft per min. Low speeds should be used for materials of large granular size such as ear corn and walnuts. Small granular material such as small grain and clover seed can be moved at higher speeds particularly if conveyed in the lower enclosed portion of the conveyor. High speeds may damage the product. Where practicable, capacity should be provided by large-size flights rather than high speeds. The capacity of a scraper conveyor operating on the level can be assumed as 115 per cent of the rectangular space between two flights when designed as suggested above. The capacity of a conveyor operating at an incline will have less capacity than when operating on the level according to the accompanying tabulation.

Incline, Degrees	*Approximate Relative Capacity*
20	0.77
30	0.55
40	0.33

The values that define the relative capacity of the conveyor will vary considerably from material to material. For example, linted cotton seed will pile much higher on a conveyor than flax seed. This will affect the relative capacity when operating either level or at an angle.

Note that the capacity of a conveyor that moves the material in the enclosed lower part of the frame would not be materially affected by the angle of incline.

The theoretical power requirement for flight conveyors can be determined from the following rational equation.

$$\text{Horsepower} = \frac{2vL_cW_cF_c + Q(LF_m + H)}{33{,}000} \tag{8.4}$$

where v = speed of conveyor, ft per min.
L_c = horizontal projected length of conveyor, ft.
W_c = weight of flights and chain, lb per ft.
F_c = coefficient of friction for chains and flights.
Q = lb material to be handled per min.
L = horizontal projected length of loaded conveyor, ft.
F_m = coefficient of friction for material.
H = height of lift, ft.

Table 8.2 FRICTION COEFFICIENTS (SLIDING)

Material		*Source*
Metal on oak	0.50–0.60	Marks
Oak on oak, parallel fibers	0.48	Marks
Oak on oak, cross fibers	0.32	Marks
Cast iron on mild steel	0.23	Marks
Mild steel on mild steel	0.57	Marks
Grain on rough board	0.30–0.45	Ketchum [6]
Grain on smooth board	0.30–0.35	Ketchum [6]
Grain on iron	0.35–0.40	Ketchum [6]
Coal on metal	0.60	Link-Belt [7]
Dry sand on metal	0.60	Link-Belt [7]
Malleable roller chain on steel	0.35	Badger and McCabe [2]
Roller-bushed chains on steel	0.20	Badger and McCabe [2]

The calculated horsepower must be adjusted to compensate for expected maximum load conditions, starting friction, loss in the driving mechanization, variation in friction coefficients, type of power unit, etc.

Example. Design a steel conveyor with the grain moving in the open top to operate at a 30° incline and elevate shelled corn to a height of 18 ft at a rate of 1500 bu per hr.

Assuming a flight speed of 100 ft per min, the cross-sectional area of the section is: 1500 bu per hr = 31.25 cu ft of grain or 1400 lb of shelled corn per min. The flight width w is:

$$w = \frac{\text{Volume material per min}}{0.4 \times \text{Relative capacity} \times \text{Flight speed}}$$

$$w = \frac{31.25}{0.4 \times 0.55 \times 100} = 1.19 \text{ ft or } 14.3 \text{ in.}$$

Use 15 in. The theoretical length or run of the conveyor would be the lift divided by the sine of the incline angle which is

$$18/\sin 30^\circ \qquad \text{or} \qquad 36 \text{ ft}$$

Additional length must be used to provide clearance and overhang at the top or discharge end. We shall add 4 ft to take care of this * making the total length 40 ft and the total actual lift 20 ft.

The load will create a tension in the chain according to the second part of equation 8.4 thus:

$$\begin{aligned} T &= \text{Wt material per ft } (L_m F_m + H) \\ &= 44.7 \times (\tfrac{15}{12})^2 \times 0.4(40 \cos 30^\circ \times 0.40 + 20) \\ &= 946 \text{ lb} \end{aligned}$$

Because of the width, two chains would be required, operating tension in each being 473 lb. A chain of 0.902-in. pitch has an ultimate strength of 1250 lb which would provide a factor of safety of over 3. This is advisable because of possible shock loads resulting from jamming. The chain weighs 0.5 lb per lineal foot and the flight about 2 lb.

Note should be made that the weight of the chain and flights add to the load and should be included in the calculations of equation 8.4. However, since the weight of the chain and flights in this example is small as compared to the material to be handled and since a moderate to high safety factor is required, weight of chains and flights may be neglected in many

* The point of discharge must be sufficiently above the specified elevation height in order to provide clearance for the housing and discharge spout. This important requirement is frequently overlooked.

calculations. If the weight of the material is low and the chain and flights significant as compared to the material, it will be advisable to include them in the initial computation. Thus the power requirement is:

$$\text{Horsepower} = \frac{2 \times 100 \times 34.6 \times 3 \times 0.50 + (1500 \times 56/60)(34.6 \times 0.40 + 20)}{33{,}000}$$

$$\text{Horsepower} = 1.75$$

A 3-hp unit would probably provide a suitable factor of safety.

SCREW CONVEYORS

Screw conveyors are used to handle finely divided powders, damp, sticky, heavy viscous materials, hot substances that may be chemically active, and granular materials of all types. Because of simplicity, freedom from sharp recesses, cracks, and crevices, dust-tightness, and ease in disassembly, screw conveyors are used for moving food products such as powdered milk and peanut butter. Screw conveyors are used for batch or continuous mixing, for feeding where a fairly accurate rate is required, and for conventional conveying and elevating jobs particularly where the run is short. Flights are made of stainless steel, copper, brass, aluminum, cast iron, etc., for hot, corrosive, or mildly abrasive materials, and are hard surfaced with Stellite or similar material to resist highly abrasive materials.

Although screw conveyors are simple and relatively inexpensive, power requirements are high and single sections are limited in length.

The standard pitch screw has a pitch approximately equal to the diameter. It is used on most horizontal installations and on inclines up to 20°. Half standard pitch screws are used for inclines greater than 20°. Double- and triple-flight, variable-pitch, and stepped-diameter screws are available for moving difficult materials and controlling feed rate. Ribbon screws are used for wet or sticky substances. Special cut flight and ribbon screws are used for mixing, both singly and in connection with conveying.

Horizontal screw conveyors are usually operated in a U-shaped trough, with or without a cover, depending upon the type of service and the characteristics of the material being moved. The

screw is supported by brackets at various standard spacings. For elevating at a steep incline, a cylindrical housing is used. The tube is operated full, and no brackets are used between the ends since they would interfere with the movement of the material. The material supports and guides the flight so that satisfactory operation results if the conveyor is kept full.

Screw conveyors are well suited as feeders or metering devices under bins or hoppers. The portion of the screw under the bin or hopper is usually designed with half or third pitch, the balance of the screw full pitch. Thus the main portion of the screw operates half or third full, whereas, the metering portion of the screw operates full.

Concise formula and data are not available for individual design problems, and it is recommended that a specialist be consulted when designing and installing large screw-conveyor systems. Data that are available to assist in design are empirical. Recommendations for design of horizontal conveyors by the Link-Belt Company [7] are shown in Fig. 8.7. The capacity of a full screw conveyor as used for feeding or elevating can be estimated from the following rational equation which is accurate enough for initial estimates.

$$\text{Theoretical capacity, cu ft per hr} = \frac{(D^2 - d^2)}{36.6} \times P \times rpm \quad (8.5)$$

where D = screw diameter, in.
d = shaft diameter, in.
P = screw pitch, in. (normally equal to D).
rpm = revolutions per minute of shaft.

The actual capacity will be much less than the theoretical because of screw-housing clearance, fluid characteristics of material, screw length, head of material, and elevation or lift. When specific operating data are not available, an estimate of 50 to 60 per cent of theoretical would be advisable.

The power requirement of a screw conveyor is a function of its length, elevation, type of hanger brackets, type of flights, the viscosity or internal resistance of the material, the coefficient of friction of the material on the flights and housing, and the weight

of the material. Consideration must also be given extra power required to start a full screw and to free a jammed screw and the power required if the material has a tendency to stick to the trough sides or to ball.

Chart I

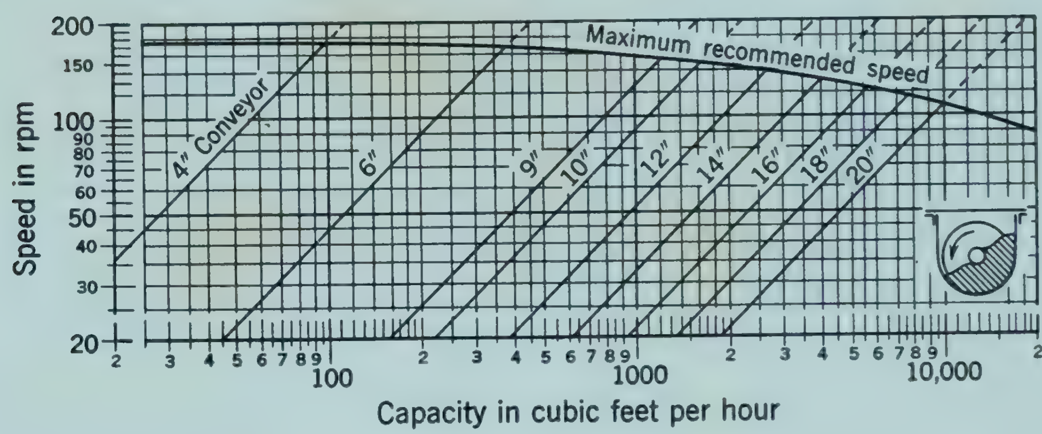


Chart II

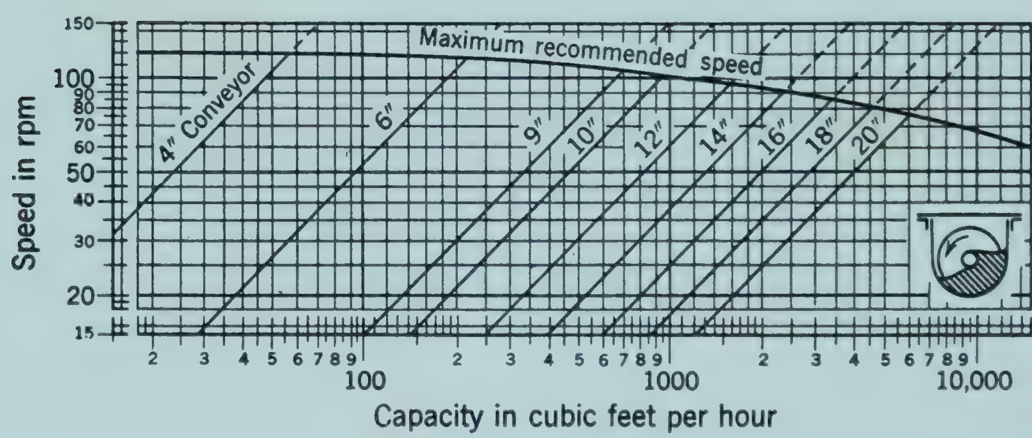


Chart III

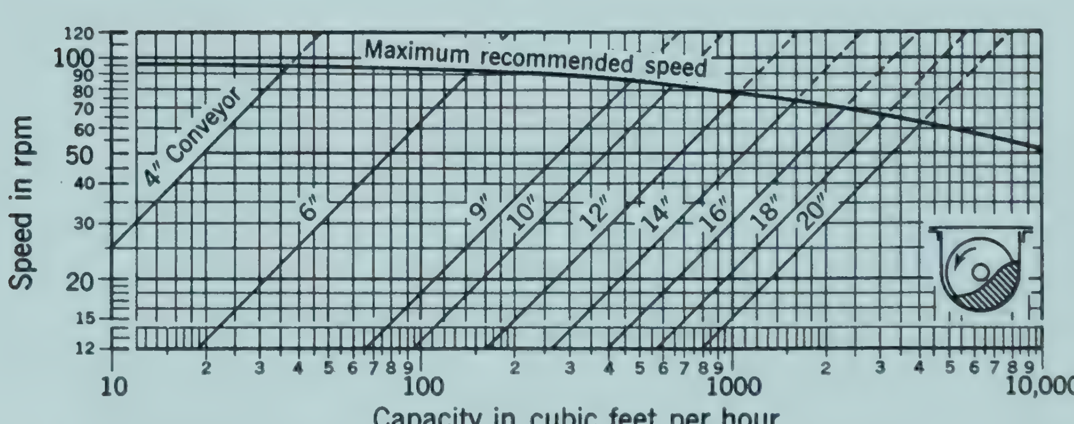


Fig. 8.7. Capacity charts for horizontal standard pitch screw conveyors. Note Table 8.3. The capacity decreases with inclination, approximately 30% for 15° and 55% for 25°. (*Courtesy* Link-Belt Co.)

Chart IV

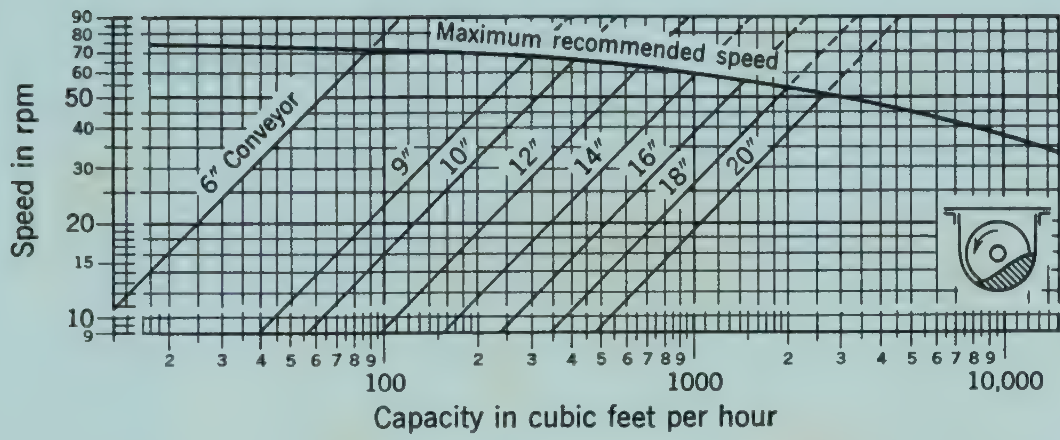


Chart V

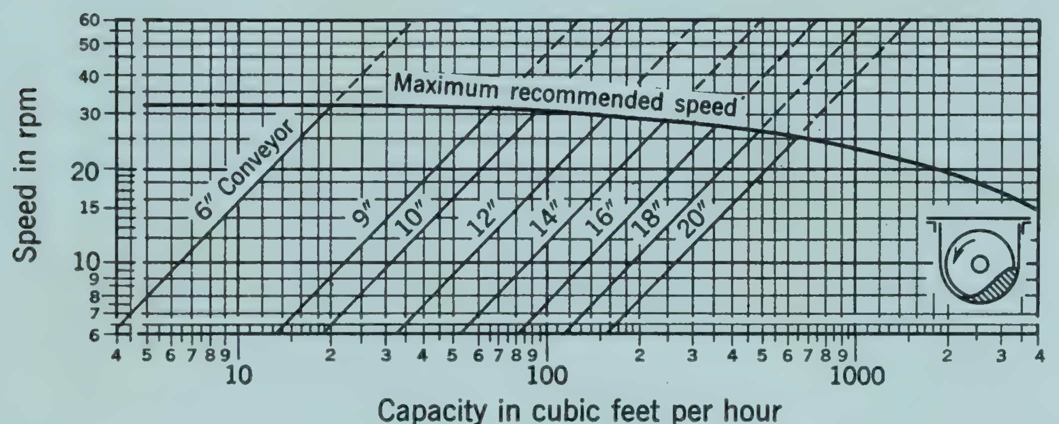


Fig. 8.7 (*continued*)

The power required to drive a screw conveyor depends upon the dimensions of the system and the characteristics of the material. An approximation for normal horizontal operation can be determined from the following equation.

$$\text{Horsepower} = CLWF/33{,}000 \tag{8.6}$$

where C = conveyor capacity, cu ft per min.
L = conveyor length, ft.
W = bulk material weight, lb per cu ft.
F = material factor, Table 8.3.

If horsepower is less than 1, double the horsepower; if 1 to 2, multiply by 1.5; if 2 to 4, multiply by 1.25; if 4 to 5, multiply by 1.1. No correction is necessary for values above 5 hp.

Table 8.3 MATERIAL CLASSIFICATION AND INDICES FOR SCREW CONVEYORS

Material	*Bulk Weight, lb per cu ft*	*Chart Number*	*Horsepower Material Factor, F*
Barley	38	I	0.4
Beans	48	II	0.4
Beans, castor	36	IV	0.5
Beans, soy	45–50	II	0.5
Bran	16	I	0.4
Butter	59	III	0.4
Corn, shelled	45	II	0.4
Cornmeal	40	I	0.4
Cotton seed (dry)	25	III	0.9
Cotton seed hulls	12	II	0.9
Lime, ground	60	III	0.6
Milk, dried	36	III	1.0
Oats	26	I	0.4
Peanuts, unshelled	15–20	IV	0.7
Rice, clean	45–48	I	0.4
Rye	44	I	0.4
Sawdust	13	III	0.7
Wheat	48	I	0.4

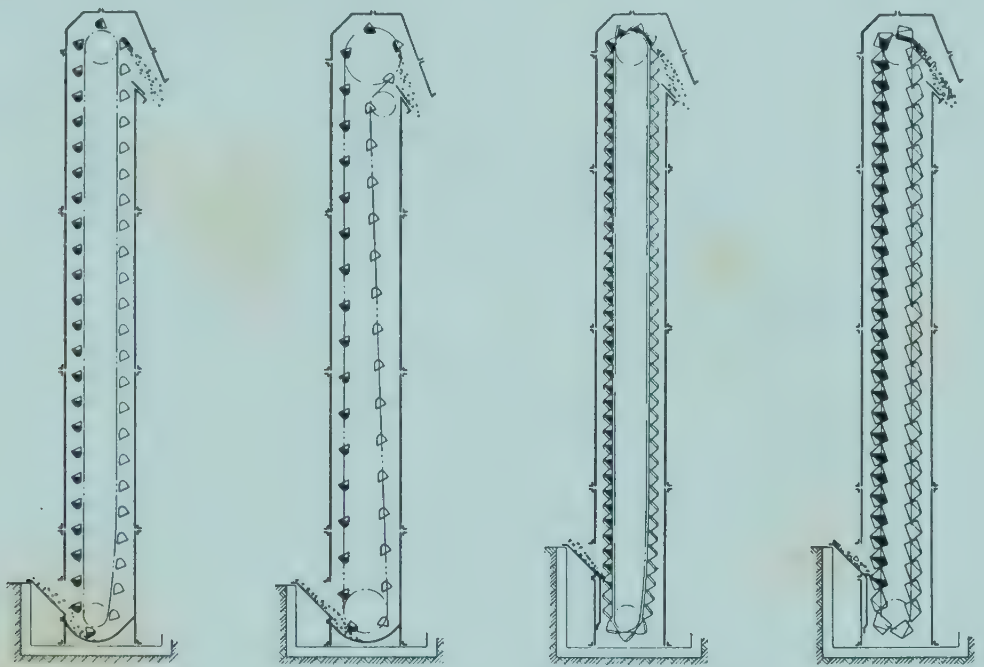
BUCKET ELEVATORS

Bucket elevators might be classed under either belt or chain conveyors, or both, since they are special adaptations of these. The adaptations are varied and range from the simple small-capacity unit used in connection with grain-cleaning equipment to the large, expensive units used for grain, coal, ashes, etc., in large industrial plants.

Bucket elevators are very efficient and are more expensive than scraper conveyors. Efficiency results from the absence of frictional loss from sliding of the material on the housing. It is this feature that distinguishes the cup elevator from the vertical or nearly vertical scraper conveyor.

Some characteristic bucket elevators are shown schematically in Fig. 8.8. The buckets may be enclosed in a single housing called a leg, or two legs may be used. The return leg may be located some distance from the elevator leg. A single or double chain or belt is used to carry the buckets. The buckets are shaped with either sharp or rounded bottoms to facilitate discharge (sect.

8.8). They are fastened to the belt or chain at the back (Fig. 8.8*a*) or at the side if two chains are used (Fig. 8.8*b*). Guides are sometimes used for two-chain installations, particularly in the up leg. Single chains and belt installations have no guides or supports between the head and foot wheels except, perhaps, an



(*a*) (*b*) (*c*) (*d*)

Fig. 8.8. Three common types of bucket elevators. (*a*) Centrifugal-discharge type; used extensively for handling small grain in elevators and processing plants. The buckets are fastened to a belt. (*b*) Perfect-discharge type. The buckets are usually fastened to a chain and operate at slow speed; they are used for materials which might be damaged or would not stay in high-speed buckets and for farms where initial cost must be low. (*c, d*) Continuous-bucket type for heavy duty use, ores, sand, etc. Material discharges by sliding over bottom of preceding bucket.

idler or two, which are placed at strategic points to eliminate whip.

8.8. Discharge from Bucket Elevators. Discharge at the head or top of a bucket elevator is produced in three different ways as shown in Fig. 8.8. Except for the overlapping buckets, which are not used extensively in processing, discharge depends upon centrifugal force in part or in full or the ability of the material to be thrown into a chute as the buckets go over the head pulley. The characteristics of this feature and, in particular, the trajectory of the material after it leaves the bucket are important to proper design and operation. Centrifugal discharge

requires that the speed of the belt or chain be held within close limits in order that the trajectory will fall within the specified region. An analysis of this follows.

Fig. 8.9 shows a head wheel and a bucket in a series of positions. A unit mass of grain is subjected to two forces at the point the bucket starts to turn around the pulley. These forces are the

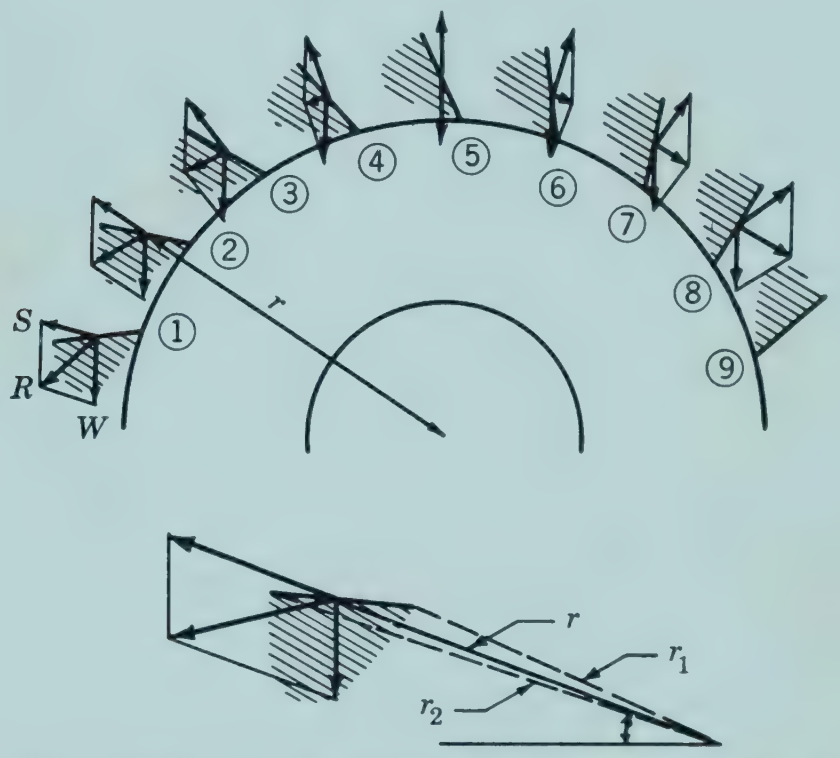


Fig. 8.9. Force diagram on the grains in a head-wheel bucket in a number of different positions. The effective radius of the head wheel bucket varies from r_1 to r_2.

weight of the unit volume W and the centrifugal force S acting radially which is

$$S = WV^2/3600gr \tag{8.7}$$

where W = weight of elemental mass, lb.
V = tangential velocity, ft per min.
g = acceleration of gravity.
r = effective radius, ft.

The resultant of these forces R, Fig. 8.9, determines the point at which discharge takes place and its characteristics.

Note in Fig. 8.9 that R for positions 1–4 is of such a direction that the material is held in the bucket. At position 5, S and W are opposing and R is zero, there being no force on the material.

Discharge begins at this point, the initial velocity and trajectory direction being that of the projected speed of the wheel at this point. Note that R in positions 6–8 is nearly in the direction of motion of the bucket and thus forces discharge.

In order to produce this condition, S and W must be equal at a point near the top of the travel, or

$$S = W = WV^2/3600gr \tag{8.8}$$

so that

$$V^2 = 3600gr$$

and since

$$V = 2\pi rN$$

Where N = revolutions per minute, then

$$N = 54.19(1/\sqrt{r}) \tag{8.9}$$

This equation shows the relationship between the effective head-wheel radius and its revolutions per minute for the most satisfactory discharge conditions. Discharge is not uniform or instantaneous because the effective radius varies from r_1 to r_2 as shown in Fig. 8.9. Thus, the material at the outer edge of the bucket discharges first.

The chute should be so placed and at such an angle that all the material will be taken and at an angle that will minimize deflection. The trajectory can be determined by the following procedure. The material is discharged in the direction of vector R which does not have to be horizontal as indicated in Fig. 8.9. The horizontal increments vary as vt, t being in small units such as 0.05 sec. The corresponding vertical distances h are those of free-falling flight or

$$h = gt^2/2 \tag{8.10}$$

By plotting the values to scale, the approximate line of flight of a discharged load can be found.

The method of pick up in the cup elevator is shown in Fig. 8.8. The foot wheel should not be much smaller than the head wheel if the speed of the head wheel conforms to the proper discharge speed. If the foot wheel is too small, the centrifugal force will not let the buckets fill. If it is desirable to use a small foot wheel a velocity somewhat under that for perfect discharge but which

will still provide satisfactory discharge should be used. The feed should be so designed that some filling will result after the buckets have passed above the center of the foot wheel. The arrangement shown in Fig. 8.8*c* is the most satisfactory since only a small portion of the material gets into the boot proper, most of it being picked up directly from the chute. Note also that if the elevator stops because of power failure or for some other reason, the influx of material into the boot is blocked. Consequently, the elevator can be started without having to clean out the boot.

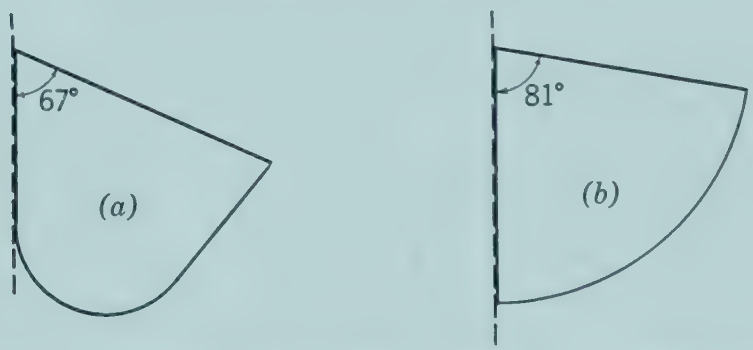


Fig. 8.10. Two representative buckets. Both buckets are available in a range of sizes. Bucket *a* has a volume of 0.104 cu ft if 10 in. long, 5⅞ in. deep (over-all) and projects 5½ in. Bucket *b* has a volume of 0.106 cu ft if 11 in. long and the curve radius is 6 in.

The elevator buckets vary somewhat in shape and size but conform in general to the two shown in Fig. 8.10. The top angle and the rounded bottom facilitate filling and discharge. The individual manufacturers furnish complete data on the various buckets which they manufacture. These data should be consulted for a specific job.

The center spacing of buckets varies with their size, shape, speed, and head and foot wheel diameter and is specified by the manufacturers for the various operating conditions. The buckets must be placed so that the centrifugal discharging grain does not hit the bucket ahead of the one discharging. In general, the spacing will be from 2.0 to 3.0 times the projected width.

The theoretical power requirement is:

$$\text{Horsepower} = QH/33{,}000 \tag{8.11}$$

where Q = amount of material handled per minute, lb.
= belt speed in feet per minute times number of buckets per foot times capacity of bucket in pounds.
H = lift, ft.

The theoretical horsepower should be increased 10–15 per cent to provide for friction and power required for loading. Also, if starting under load or if heavy peak load conditions are frequently expected, additional power should be provided.

Take-up for chain-wear and belt-tension adjustment are usually provided by moving the foot pulley. The power should be applied at the head pulley.

PNEUMATIC CONVEYORS

The pneumatic conveyor moves granular material in a closed duct by a high-velocity stream of air by systems such as shown

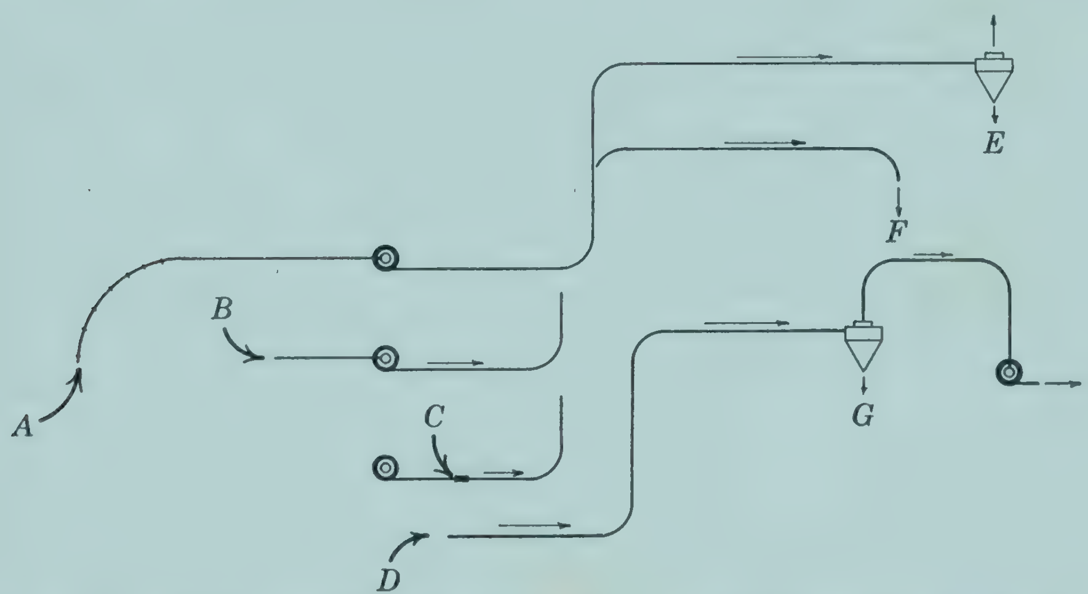


Fig. 8.11. Pneumatic conveying systems. The material may be sucked up by a flexible hose, *A*, may be introduced into the intake pipe or directly into the fan by gravity flow or a hopper, *B*, or ahead of the fan, *C*. The material may be collected by a cyclone, *E*, or discharged directly, *F*.

in Fig. 8.11. The advantages are: relatively low initial cost; mechanical simplicity (only one major moving part, the fan); conveying path can be random and may branch; conveying path can be changed easily; a wide variety of materials can be conveyed (dusts, fibers, sand, grain, rags, cotton, etc.), and the system is self-cleaning. The disadvantages are: high power requirement and possible damage to conveyed materials.

Conveying engineers are not in complete agreement as to the performance and design aspects of pneumatic conveying. The following treatment prepared from various sources[2,5,9,11] is believed by the authors to be adequate for average design.

8.9. Systems. Three systems may be listed. (1) *Suction* systems, Fig. 8.11*d*, operate below atmospheric pressure. (2) *Low-pressure* systems, Fig. 8.11*c*, use high-velocity low-density air. The system is usually powered by a centrifugal fan since the operating pressure is low to moderate, up to approximately 14 in.

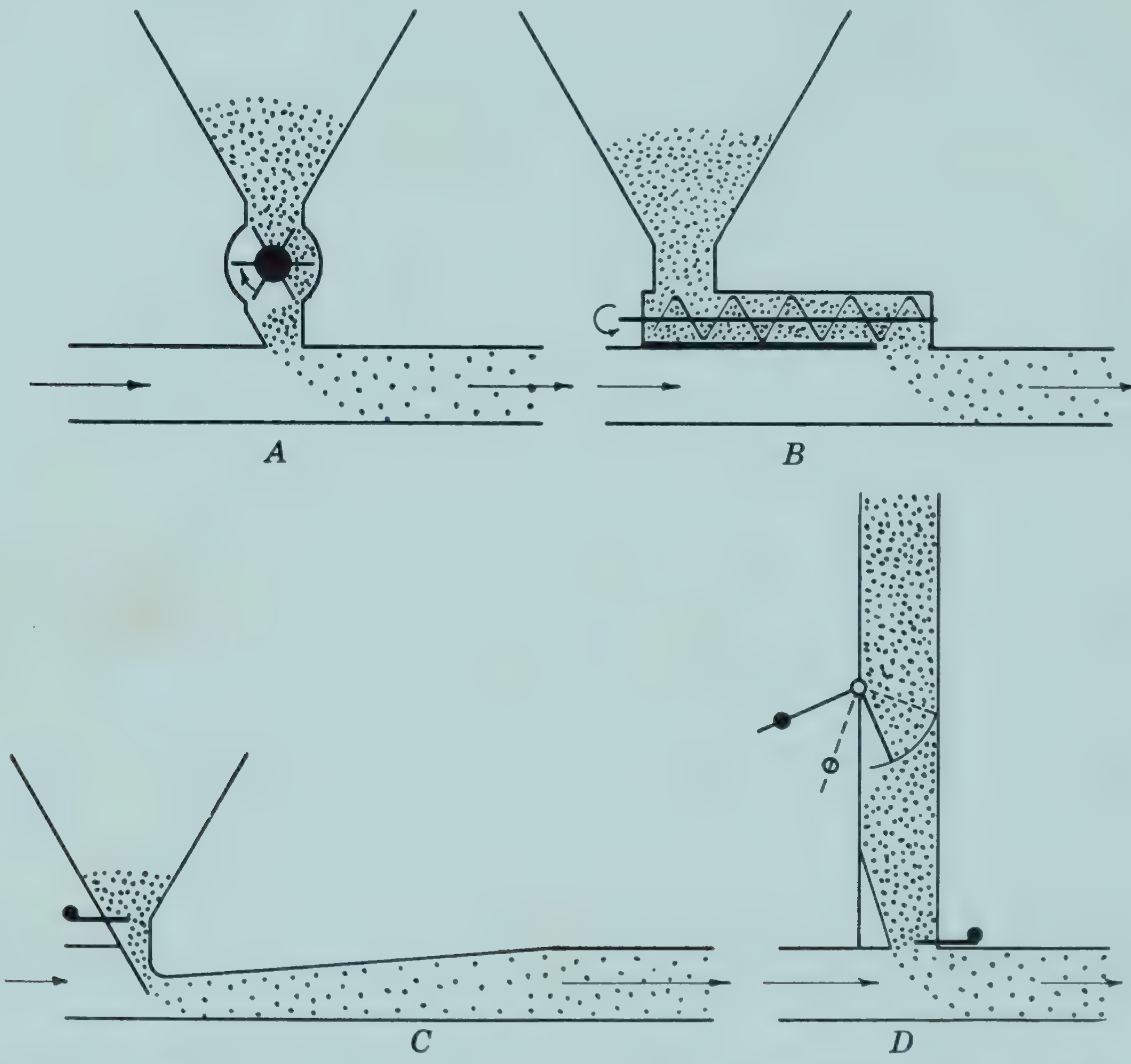


Fig. 8.12. Material-feeding devices for system *C*, Fig. 8.11. Bucket wheel *A*, auger *B*, injector *C*, and column *D*. The column gate closes as the material surface drops, thus restricting the back flow of air.

of water. (3) *High-pressure* systems use low-velocity high-density air. Positive displacement blowers are usually required for high-pressure systems. System *D–G* is a true suction system since the pressure is less than atmospheric through the entire conveying distance. Although systems *A* and *B* are classed as pressure systems, they are actually combination systems since pressures vary from below to above atmospheric.

The suction system is best for unloading materials where the point of unloading may move or there are a number of locations

from which material is to be taken. Unloading from trucks, wagons, freight cars, and boats are examples. The suction system is also best for materials of such a texture that they would not pass easily through valves, screw feeders, or fans, cotton, for example. The pressure system is more efficient than the suction system since the conveying air density is higher and velocity lower. It is best adapted for work where the point of discharge varies. Loading freight cars or storage tanks are examples.

The method or device used to feed the material into the air stream is important owing to the effect on the material itself and the power required. In addition to the devices of Fig. 8.12 the material may be introduced into the fan inlet or sucked into the intake as in Fig. 8.11*A*, *B*. The possibility of fan damage must be recognized in the last two procedures.

8.10. Air Rate and Volume. An accepted material conveying rate is 50 ft per sec. For vertical movement the air velocity must be that to just support the particles plus 50 ft per sec. A higher material velocity is needed for horizontal conveying since high turbulence is required to maintain the material in suspension. The air rate may be less since the air-particle slip is less for horizontal movement. The proper air rate for lifting can be calculated by the aerodynamic procedure of sect. 7.12 or by empirical methods such as those of Hudson.[5] Hudson's recommendations, based upon the observed performance of many installations, include velocities for horizontal conveying and flexible pickup hoses and are given in equation 8.12 and Table 8.4.

$$V = 60M_c\sqrt{W} \tag{8.12}$$

where V = air velocity, ft per min.
M_c = a constant, Table 8.4.
W = bulk density, lb per cu ft.

Table 8.4 VALUES OF M_c FOR EQUATION 8.12

	Straight Horizontal Ducts		*Ducts with Ells and Risers*	
Material	*Line*	*Hose*	*Line*	*Hose*
Dusty	10	16	12.50	20
Grains	12	20	15	24
Gritty and uneven	15	24	18.75	30

Some conveying velocities recommended by the Buffalo Forge Company[9] are noted in Table 8.5.

Table 8.5 RECOMMENDED CONVEYING AIR VELOCITIES BY THE BUFFALO FORGE CO., FT PER MIN

Castor beans	5000	Sand	7000
Corn	5600	Sawdust	3000
Cotton	4500	Shavings	3500
Oats	4500	Wheat	5800 29.4 mts/sec
Paper	5000	Wool	5000
Rags	4500	Vegetable pulp, dry	4500

The quantity of material conveyed per cubic foot of air depends upon the operating pressure and uniformity of feed. Under normal low-pressure system operation 1 lb of material may be handled by each 35 to 50 cu ft of air. High air rates permit a higher concentration of material. High-pressure systems will carry more material per cubic foot of air due to the increased air density.

The conveying pipe diameter for a desired material rate can be calculated from the data presented above by selecting appropriate values of air rate and material air ratio.

8.11. Feeding Devices. The feeding device is important as regards (1) damage to the material conveyed, (2) power required for operation, (3) initial cost and performance. The material may enter the conveying system by being (1) introduced directly into the fan (Fig. 8.11*B*), (2) sucked up by flexible hose (Fig. 8.11*A*) or (3) metered into the moving air stream by (*a*) a bucketwheel, (*b*) an auger, (*c*) an injector, or (*d*) a column feeder (Fig. 8.12).

The simplest, most efficient, and cheapest feed is directly into the fan. However, many materials may be damaged by this procedure. The suction hose system is convenient, but the hose velocity must be high. The bucketwheel and auger meters provide a uniform flow of material without introduction damage, but they are expensive and some back air leakage takes place through the meter. This loss is usually insignificant. The column meter is simple, but back leakage may be significant at low column heights. The venturi meter reduces the pressure to atmospheric (or lower) at the point of material entrance. It is not as ex-

pensive as the mechanical feeders, even though it necessitates added power at the fan.

8.12. Operating Pressure. The operating or fan pressure differential is composed of (1) air-pressure loss in pipe and fittings, (2) material-pipe friction, (3) material-air friction, (4) material acceleration, and (5) sundries such as occurs in cyclones, valves, material meters, etc. The total pressure is the sum of the individual pressure differentials from the sources noted.

1. The pressure loss in the *pipe* and *fittings* can be calculated by any conventional method such as that of Chap. 2. Friction, elbow, and velocity losses must be included. The effect of material collectors, venturi or other feeding devices, transfer valves, and other sundries upon the pressure loss must be included in the calculations.

2. The mechanics of the loss due to *material* moving in the pipe are not well known. The friction of the material against the pipe is probably significant for horizontal movement but not so important for vertical movement. Aerodynamic loss is probably high for vertical movement, but pipe-material friction is probably of little importance for horizontal movement.

The Buffalo Forge Company [9] uses the following expressions developed from tests by M. Gasterstädt [3] to determine the pressure loss resulting from the presence of the material when moved horizontally.

$$m = a\left(\frac{R}{K} + 1\right) \tag{8.13}$$

where m = total pressure drop through a system handling material in horizontal pipes.

a = pressure drop through the system handling air only.

R = ratio, lb material per lb air.

K = a variable that depends upon velocity as noted below.

Velocity, ft per min	2000	3000	4000	5000	6000
K	1.15	2.14	3.11	3.5	3.5

The resistance characteristics for vertical movement are not well known. The Buffalo Forge Company [9] uses a vertical resistance twice that for horizontal movement for moderate lifts (approximately 50 ft or less).

3. Energy loss in *elbows* can be estimated by the equivalent length of vertical pipe which is then applied to equation 8.13.

4. The energy required to *accelerate* the material from zero to the conveying velocity is significant. The pressure loss resulting from the introduction of the material into the stream of moving air may be estimated by the following expression which was developed by Gasterstädt.

$$m = 2.25Rp \tag{8.14}$$

where m = pressure loss, in. water.
R = ratio, lb material per lb air.
p = velocity pressure, moving air.

GRAVITY CONVEYORS

Gravity conveyors consist of a series of rollers or wheels set level or at a slight incline to handle boxed materials in particular. The material moves by gravity, or by hand if the conveyor is level. It is especially adapted for intermittent service.

CRANES

Outside of the conventional rope hay hoist used for stacking and barn filling, cranes are not used extensively in agricultural processing. They may be used in isolated instances, but their use is not general enough to warrant more than this passing comment.

LIFT AND CARRYING TRUCKS AND CARTS

Much of the processing done on farm products is concentrated into a relatively short period during the year. The critical element in the flow of materials through a processing plant is usually the peak of receipt of raw material. Vegetables and fruits have to be delivered at the most optimum time. The daily peak rate of receipt is usually higher than the maximum processing rate. Consequently, the material may have to be unloaded and stored temporarily before starting through the plant.

Furthermore, many plants operate only a portion of the year and consequently have to minimize overhead as much as possible. Rather than buy an expensive machine for a certain unit opera-

tion that is performed only a few days out of the year, it may be economically advisable to use a slower process taking a longer time. This in turn would require a change in the rate of movement through the plant which would probably require temporary interruption or storage of material. Consequently, conventional continuous-flow procedures would not apply satisfactorily.

The processing characteristics of any one plant vary from year to year and from day to day. For example, incoming apples on a certain day may grade out 10 per cent for cider. The following day apples from another source may grade 25 per cent cider. If the plant does not have a cider mill of sufficient capacity to handle the quantity at the rate received, temporary holding will be necessary and normal-flow procedures will be disrupted.

Because of these unique aspects of agricultural processing, that is, possible day-to-day variations in quantity and quality of material received, the necessity for a minimum overhead, and variation in direction and rate of flow through the plant, a complete permanent conveyor system may be inadvisable. This does not preclude the advisability of permanent installations in many plants or connecting most of the operations in any one plant. The most suitable system can be determined only after a careful overall study of the enterprise.*

Carts, dollies, and trucks may be satisfactory for handling material under these conditions. Labor efficiency is possible only when these devices are fitted to the work in the most satisfactory manner. Lifting by the operator should be minimized, the cart or truck should fit the material to be moved so that each load will be balanced for easy movement and a maximum load will be carried each trip. If the material is handled in boxes, efficiency is provided if the boxes nest so that the resultant load moves as a unit.

For moderate, large-size operations with a high annual use factor, a lift truck with pallets designed to handle the material is a fast and efficient device for materials handling where the speed and direction of movement varies from day to day. Pallets are designed in various ways and may be platforms, racks, boxes, etc. The chief feature to consider is the method of attachment to the truck.

* Note Chapter 14.

The lift-truck-pallet system of materials handling is very efficient for certain phases of many agricultural processing operations. It should be considered for use where the flow rate is intermittent, and where bags, bales, hampers, boxes, and similar large units are handled.

REFERENCES

1. Alden, J. L. *Design of Industrial Exhaust Systems.* Industrial Press. 1939.
2. Badger, W. L., and McCabe, W. L. *Elements of Chemical Engineering.* Second Edition. McGraw-Hill. 1936.
3. Gasterstädt, M. Experimental Investigation of Pneumatic Conveying Process. *Zeitschrift des Vereines Deutscher Ingenieure. 68.* Pp. 617–624. June 14, 1924.
4. Hetzel, F. V., and Albright, R. K. *Belt Conveyors and Belt Elevators.* Third Edition. Wiley. 1941.
5. Hudson, W. G. *Conveyors and Related Equipment.* Wiley. 1944.
6. Ketchum, M. S. *The Design of Walls, Bins, and Grain Elevators.* Third Edition. McGraw-Hill. 1919.
7. Link-Belt General Catalogue 800. Link-Belt Co. 1939.
8. Longhouse, A. D., et al. The Application of Fluidization to Conveying Grain. A Progress Report. *Agr. Eng.* 31:349, 352. 1950.
9. Madison, R. D. *Fan Engineering.* Fifth Edition. Buffalo Forge Co. 1948.
10. Rex Conveyors and Process Equipment. Catalogue 445. Chain Belt Co. 1945.
11. Segler, G. *Pneumatic Grain Conveying with Special Reference to Agricultural Application.* National Institute of Agricultural Engineering. Wrest Park, Silsoe, Bedfordshire, England. 1951.

PROBLEMS

1. Shelled dent corn is to be conveyed 40 ft horizontally and lifted 5 ft vertically at a rate of 1200 bu per hr.

   a. Specify a screw conveyor, its speed and horsepower.
   b. Specify a steel scraper flight conveyor, its speed and horsepower. Two chains are used; chain weighs ⅓ lb per ft. Flights are $\frac{3}{16}$ in. thick and travel 100 ft per min. What percentage of the required power is due to the flight and chain friction?
   c. What power would be required in (*b*) if roller chain, Fig. 8.4, with a coefficient to friction of 0.06 were used? What is the reduction in per cent?

2. Wheat is to be moved 400 ft horizontally by a belt conveyor at a rate of 3000 bu per hr.

a. Specify the belt width, surcharge angle, belt speed, and power required.
b. How much power is absorbed in accelerating the grain? The grain is moving 90 ft per min horizontally when it hits the belt.

3. Buckets geometrically similar to Fig. 8.10*a* are 12 in. long and are spaced 3.0 times the projection. The lift is 80 ft. The head wheel is 24 in. in diameter. Determine:

   a. Belt speed for centrifugal discharge.
   b. Capacity in bushels per hour.
   c. Horsepower when elevating shelled corn.

4. Plot the discharge trajectory for problem 3 for r_1 and r_2 (Fig. 8.9). Does the grain completely clear the discharged buckets? Sketch the outline of the discharge chute.
5. A pneumatic conveyor must lift wheat 35 ft and move it horizontally 60 ft at 500 bu per hr. The system is *A–F* of Fig. 8.11. The hose is 10 ft long. The radius of the ells is 6 times the pipe diameter. Determine:

   a. Hose and pipe diameter (use 40 cu ft air per pound of material).
   b. Pressure drop across the fan.
   c. Air horsepower.

6. Determine the horsepower for a scraper conveyor for the job of problem 5. Assume steel construction; flight and chain weight, 2 lb per lineal foot; flight speed, 100 ft per min.

CHAPTER 9

Heat Transfer

NOMENCLATURE

A = cross-sectional area, sq ft.
C = a constant.
c = specific heat, Btu per lb °F.
D = diameter, ft.
G = $V\gamma$ = mass rate, lb per (hr sq ft).
h = surface thermal conductance, Btu per (sq ft hr °F).
I = radiant-energy rate, Btu per (hr sq ft steradians).
k = thermal conductivity, Btu per (hr sq ft °F per ft).
L = thickness of a conducting layer, ft.
N = cylinder length, ft.
q = heat rate, Btu per hr.
Re = Reynolds number.
r = radius, ft.
T = absolute temperature, °R.
t = temperature, °F.
t_{av} = average temperature, °F.
t_c = center temperature, °F.
t_f = fluid film temperature, °F.
t_s = surface temperature, °F.
U = over-all or composite conductivity, Btu per (hr sq ft °F).
V = velocity, ft per sec.
v = volume, cu ft.
W = total emissive power, Btu per (hr sq ft).
W_b = black-body emissive power, Btu per (hr sq ft).
$W_{b\lambda}$ = spectral emissive power, Btu per (hr sq ft micron).
w = fluid weight, lb per hr.
x = distance through conducting medium, ft.
x, y, z = coordinate axis, ft.
α = absorptivity, a ratio.
α_λ = monochromatic absorptivity, a ratio.
β = coefficient of cubical expansion, cu ft per cu ft °F.
Γ = total irradiation, Btu per (hr sq ft).
Γ_λ = monochromatic irradiation, Btu per (hr sq ft micron).
ϵ = emissivity, a ratio.
ϵ_λ = monochromatic emissivity, a ratio.
γ = specific weight, lb per cu ft.
θ = time, hr.

λ = wave length, microns (one micron = 0.001 mm).
μ = viscosity, lb per ft-sec.
ρ = reflectivity, a ratio.
ρ_λ = monochromatic reflectivity, a ratio.
σ = a constant, 0.173×10^{-8} Btu per (hr sq ft °F^4).
ϕ = angle, degrees.
ψ = a function of.
ω = solid angle, steradians.

Transfer of heat is the principal unit operation in the processing of many farm products, for example, pasteurization of milk and fruit juices, freezing of foods, cooling of fruits and vegetables for transportation and storage, and thermal sterilization of canned foods. Heat transfer is also an essential operation in providing the energy for vaporization in evaporation, distillation, and drying. Heat must be supplied to maintain desirable temperatures for bacterial growth in cottage-cheese making; on the other hand, it must be removed to avoid undesirably high temperatures in fermentation processes.

Heat energy is transferred by three mechanisms: conduction, convection, and radiation. In many systems, all three operate simultaneously. Superficial consideration of such systems has often yielded rule-of-thumb expressions that are obviously simple, but which have a limited range of application. Sound treatment requires a recognition of the part played by each pertinent mechanism. The range where simplifications are valid can then be established.

9.1. Conduction. Transfer of heat energy between adjacent molecules, not dependent on gross movement of material, is called conduction. Substances have the ability to conduct heat in any state, solid, liquid and gaseous. In experiments to measure conduction properties of fluids, special care must be taken to avoid gross movement.

The rate of heat transfer by conduction through a substance is directly proportional to the temperature gradient, dt/dx, and to the cross-sectional area of the path, thus:

$$q = -kA(dt/dx) \tag{9.1}$$

where q = heat rate, Btu per hr.
A = cross-sectional area of flow path, sq ft.
t = temperature, °F.
x = distance through conducting medium, ft.

Table 9.1 COEFFICIENTS OF THERMAL CONDUCTIVITY OF VARIOUS MATERIALS *

Btu per (hr sq ft °F per ft)

Material	*Apparent Density, lb per cu ft*	*°F*	*k*
Air		32	0.0140
Asbestos, cement boards	120	68	0.43
Asbestos sheets	55.5	124	0.096
Asbestos	36	32	0.090
	36	212	0.111
Aluminum		32	117
Aluminum foil, 7 air spaces per 2.5 in.	0.2	100	0.025
Brick, building		68	0.4
Cardboard, corrugated			0.037
Concrete 1:4 dry			0.44
Concrete, stone			0.54
Copper, pure		64	224
		212	218
Cotton wool	5	86	0.024
Cork, board	10	86	0.025
Cork, ground	9.4	86	0.025
Diatomaceous earth	27.7	399	0.066
	27.7	1600	0.092
Fiber insulating board	14.8	70	0.028
Glass, boro-silicate	139	86–167	0.63
Glass, soda			0.3–0.44
Glass, window			0.3–0.61
Ice	57.5	32	1.3
Iron, wrought		64	34.9
		212	34.6
Iron, cast		129	27.6
		216	26.8
Mill shavings			0.033–0.05
Mineral wool	9.4	86	0.0225
	19.7	86	0.024
Sawdust	12	70	0.03
Snow	34.7	32	0.27
Steel, mild		64	26.2
		212	25.9
Steel, stainless (18-8)		932	12.4
Water		32	0.330
Wood shavings	8.8	86	0.034
Wood, across grain, balsa	7–8	86	0.025–0.3
oak	51.5	59	0.12
white pine	34	59	0.087
Wool, animal	6.9	86	0.021

* From McAdams [12] and other sources.

In equation 9.1, the proportionality constant k, called the thermal conductivity, is a property of the conducting material. Values for common materials are given in Table 9.1 where k is expressed in Btu per (hr sq ft °F per ft).

9.2. Convention. Transfer of heat by transport of heated fluid material is convection. The transport may be (*a*) natural or free convection, caused by difference in buoyancy, or (*b*) forced convection, accomplished mechanically with pumps, blowers, or fans. The principal resistance to heat transfer is found in a relatively stagnant laminar layer and an adjacent turbulent zone of fluid at the solid-fluid interface. Heat must pass through the laminar layer by conduction in the fluid. The heat rate is proportional to the difference in temperature between the surface and the main bulk of fluid and to the surface area, thus:

$$q = h_c A(t_s - t_f) \tag{9.2}$$

In equation 9.2, the proportionality constant h_c is called the unit-surface thermal conductance for convection (popularly the heat-transfer coefficient). It is determined by the properties of the fluid, the nature of the surface, and the manner and velocity of the fluid flow past the surface. It can be regarded as the conductance k/x_f of a layer of the fluid of fictitious thickness x_f through which heat can pass only by conduction. Representative values of surface conductances are given in Table 9.2.

Table 9.2 REPRESENTATIVE SURFACE-CONDUCTANCE COEFFICIENTS

Service	h_c *Btu per (sq ft hr °F)*
Evaporating water	400–4000
Condensing steam	300–5000
Evaporating ammonia	300–500
Condensing ammonia	900–1600
Air on wall surface, natural convection	1.65 *
Air on wall surface, 15-mph wind	6.00 *
Air forced across 1-in. tubes at 10 ft per sec	7
Water at 4 ft per sec in 1-in. pipe	930
Surface cooler, milk flowing over horizontal tubes	200–650

* Radiant transfer is included in these values.

9.3. Radiation. Radiation is the emission of energy, without need of a conducting or convecting medium, from the surfaces of opaque bodies and from within semitransparent objects. Fundamentally, radiant phenomena are described partly by electromagnetic wave theory and partly by quantum theory. A narrow band of thermal radiation stimulates the eye and is called light. For example, energy is transferred from the sun to the earth by thermal radiation. However, thermal radiation is also emitted at low temperatures, thus the human body dissipates heat partly by radiation. Radiation rate depends upon the area and the nature and absolute temperature of the surface. The relationship of these factors is defined thus:

$$q = A\epsilon\sigma T^4 \tag{9.3}$$

The first proportionality constant, ϵ, the emissivity, is a property of the surface of material, the ratio of its emissive power to that of a perfect radiator. The second, σ, the Stefan-Boltzmann constant, is a numerical constant for a perfect radiator, 0.173×10^{-8} Btu per (hr sq ft °R^4). With this numerical value, the temperature must be given in degrees Rankine, i.e., degrees Fahrenheit absolute or (°F + 460).

CONDUCTION

The thermal conductivity of most materials is a function of the temperature. Note Table 9.1 for conductivity values, some of which are given at two temperatures. That of most pure metals decreases slightly with rise in temperature. The conductivity of alloys is often lower than that of the principal constituents and usually rises with increase in temperature. For most insulating materials, excepting magnesite refractory brick, conductivity increases with temperature. When conductivity is linear with temperature, as is usually true for fairly wide ranges, the heat rate in a plane wall can be found correctly by using conductivity at the arithmetic mean of the temperatures of the faces.

9.4. Steady-State Conduction; Plane Walls. When the boundary temperatures have been constant long enough that temperatures within the system have ceased to change with time a steady-state conduction condition exists. In plane walls (pre-

suming that edges and corners are negligible or are protected to prevent lateral transfer) the flow is perpendicular to the faces, so that A does not vary with distance through the wall. Equation 9.1 can be directly integrated between definite limits to give

$$q = -kA(t_2 - t_1)/(x_2 - x_1) \tag{9.4}$$

Example. Find the rate of heat flow through a cork partition wall in a cold-storage plant. The wall is 16 ft long, 9 ft high, and 4 in. thick. The temperatures of the two faces are maintained at 31 and 3°F respectively. Inserting the dimensions and temperatures into equation 9.4 and the thermal conductivity (from Table 9.1) of 0.025 (more precisely, a mean value at the mean temperature of 17°F should be used),

$$q = 0.025(16 \times 9)\frac{(31 - 3)}{(\frac{4}{12} - 0)} = 302.4 \text{ Btu per hr}$$

In composite plane walls at steady state the heat rate per square foot through each component is constant and flow proceeds in series through the several materials. Equation 9.4 solved for the temperature drop through a layer gives

$$t_1 - t_2 = q(x_2 - x_1)/kA \tag{9.5}$$

Equation 9.5 is analogous to Ohm's law; the temperature drop is the product of the heat rate and the resistance. It can be seen here, also, that resistance is the reciprocal of conductance. Similarly, for convective transfer, the surface resistance can be found from equation 9.2 to be $1/h_cA$. For the composite wall shown in Fig. 9.1,

$$t_i - t_0 = (t_i - t_1) + (t_1 - t_2) + (t_2 - t_3) + (t_3 - t_4) + (t_4 - t_0) \tag{9.6}$$

$$= q\left[\left(\frac{1}{h_iA}\right) + \left(\frac{x_2 - x_1}{k_aA}\right) + \left(\frac{x_3 - x_2}{k_bA}\right) + \left(\frac{x_4 - x_3}{k_cA}\right) + \left(\frac{1}{h_0A}\right)\right] \tag{9.7}$$

Solving for q,

$$q = \frac{(t_i - t_0)}{\dfrac{1}{h_iA} + \dfrac{L_a}{k_aA} + \dfrac{L_b}{k_bA} + \dfrac{L_c}{k_cA} + \dfrac{1}{h_0A}} \tag{9.8}$$

where L is the thickness in feet of the layer considered. For heat transfer in plane walls, it is usually more convenient to use the

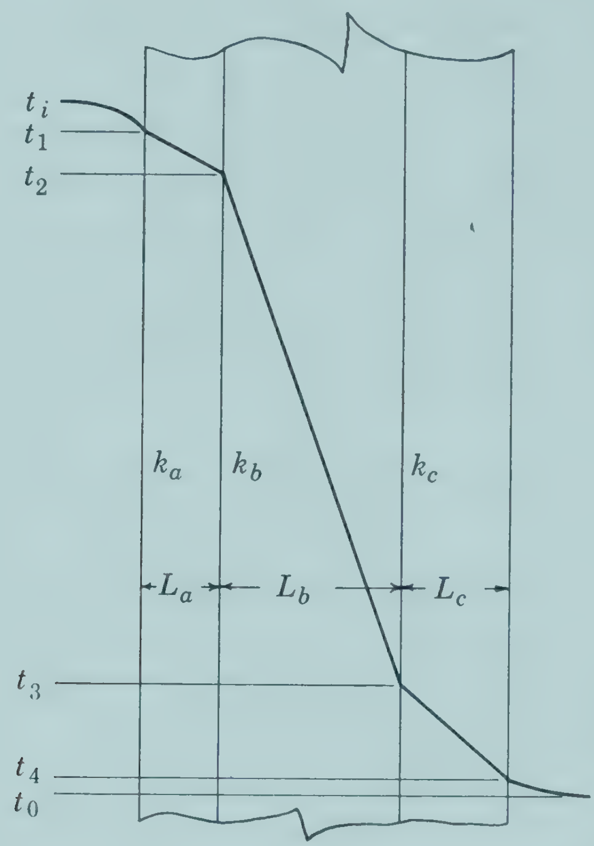


Fig. 9.1. Composite wall showing the temperature gradient.

unit resistance, factoring the area A out of each of the resistance terms, to give

$$\frac{q}{A} = \frac{(t_i - t_0)}{\dfrac{1}{h_i} + \dfrac{L_a}{k_a} + \dfrac{L_b}{k_b} + \dfrac{L_c}{k_c} + \dfrac{1}{h_0}} \qquad (9.9)$$

The heat rate is also conveniently expressed by use of an over-all unit conductance U in

$$q = UA(t_i - t_0) \qquad (9.10)$$

Inspection of equations 9.9 and 9.10 reveals that U is the reciprocal of the unit resistance given by the denominator of equation 9.9.

Example. The exterior wall of a cold room, 16 ft long and 9 ft high, is built of 6 in. of concrete, 4 in. of cork, ¾ in. of cement plaster. Find the steady-state heat rate when the outside temperature averages 80°F and the room temperature is 0°F. Also find the temperature of the concrete-cork interface.

From Table 9.1, the conductivities of concrete, cork, and plaster are 0.54, 0.025, and 0.5 Btu per (hr sq ft °F per ft) respectively. From Table 9.2, the outside surface conductance (for 15 mph wind, with radiant transfer) is 6, and the inside surface conductance (for "still" air) is 1.65 Btu per (hr sq ft °F).

$$q = \frac{16 \times 9 \times (80 - 0)}{\frac{1}{6} + \frac{\frac{6}{12}}{0.54} + \frac{\frac{4}{12}}{0.025} + \frac{\frac{3}{48}}{0.5} + \frac{1}{1.65}}$$

$$= \frac{144 \times 80}{0.167 + 0.928 + 13.333 + 0.125 + 0.605}$$

$$= 11{,}520/15.157 = 760 \text{ Btu per hr}$$

The temperature drop from the outside to the concrete-cork interface is proportional to the resistance encountered up to that point. As given in the denominator above, the over-all resistance is 15.157°F per (Btu per hr sq ft), while that from the adjacent 80°F air to the desired interface is $0.167 + 0.928$ or 1.094. The temperature at the point is then $80 - (1.094/15.175)(80 - 0)$ or 74.2°F.

9.5. Steady-State Conduction, Cylinders. For conduction in cylindrical objects, it is convenient to write equation 9.1 in cylindrical coordinates,

$$q = -kA(dt/dr) \tag{9.11}$$

r being radius in feet.

For steady-state radial flow, with circular symmetry and negligible transfer at the ends, since $A = 2\pi rN$, N being the cylinder length in feet,

$$dt = -q/2\pi kN\, dr/r \tag{9.12}$$

Integrating equation 9.12 between definite limits t_1 and t_2 and r_1 and r_2 respectively,

$$t_1 - t_2 = q/2\pi kN \ln r_2/r_1 \tag{9.13}$$

Equation 9.13 reveals that the temperature is linear with the logarithm of the radius and that the resistance to heat flow depends upon the logarithm of the radius ratio and not simply upon the thickness.

Heat flow in the hollow cylinder can be expressed in a form similar to equation 9.4 by the use of a mean radius for finding an equivalent area for an equivalent plane wall, thus:

$$q = -\frac{2\pi r_m Nk(t_1 - t_2)}{r_1 - r_2} \tag{9.14}$$

Substitution of the equivalent of $(t_1 - t_2)$ from equation 9.13 into equation 9.14, and solving for r_m yields

$$r_m = \frac{r_2 - r_1}{\ln r_2/r_1} \tag{9.15}$$

As would be expected, a logarithmic mean is the result.

For composite cylinders, resistances can be added in a manner similar to that for composite plane walls. Resistances per lineal foot are customarily used. The expression analogous to equation 9.9 is

$$\frac{q}{2\pi N} = \frac{(t_i - t_0)}{\dfrac{1}{h_i r_1} + \dfrac{\ln r_2/r_1}{k_a} + \dfrac{\ln r_3/r_2}{k_b} + \dfrac{\ln r_4/r_3}{k_c} + \dfrac{1}{h_0 r_4}} \tag{9.16}$$

9.6. Multidimensional Heat Flow. Treatment required for heat flow in two and three dimensions is beyond the scope of this text. Analytic solutions for a number of regular shapes are available in Carslaw [3] and in Boelter, et al.[1] For irregular shapes, in two dimensions, the flux-plot method of estimating flow normal to isotherms can be used. The relaxation method of Southwell [17] has been applied both to two- and three-dimensional irregular shapes.

9.7. Transient Heat Conduction. Transient heat conduction occurs when boundary conditions change suddenly, or vary with time, so that the temperature at any given point does not remain constant. Cooling of meat, fruits, and vegetables, and thermal processing of canned foods are examples.

A simple case is that of an object of high conductivity, suddenly placed in surroundings of a different temperature t_0. If the conductivity is high compared with the surface conductance, and the size is small (specifically if hr/k is less than 0.2), there will be a negligible gradient within the object, so that the mean temperature t will at no time differ appreciably from the surface temperature. The heat energy required to raise the temperature is gained from the surroundings, thus

$$c\gamma v \, dt = hA(t_0 - t) \, d\theta \tag{9.17}$$

$$dt/(t_0 - t) = (hA/c\gamma v) \, d\theta \tag{9.18}$$

where v = volume, cu ft.
c = specific heat, Btu per lb °F.
γ = specific weight, lb per cu ft.
θ = time, hr.

Where the thermal properties do not change with temperature and the surrounding temperature is constant, equation 9.18 can be readily integrated. Using the lower limits of $t = t_1$ when $\theta = 0$ and the upper indefinite limit of $t = t$ when $\theta = \theta$ in order to find the variation of t with time,

$$\ln \frac{t - t_0}{t_1 - t_0} = \frac{-hA}{c\gamma v}\theta \tag{9.19a}$$

or

$$\frac{t - t_0}{t_1 - t_0} = e^{\frac{-hA}{c\gamma v}\theta} \tag{9.19b}$$

Example. Find the temperature of a steel bar, 3 in. in diameter and 12 in. long, initially at 70°F, 40 min after being suddenly placed in an annealing furnace where the temperature is 600°F, if the surface thermal conductance is estimated to be 6 Btu per (hr sq ft °F). The specific heat of steel is 0.12 Btu per lb °F, the specific weight 450 lb per cu ft, and the thermal conductivity 26 Btu per (hr sq ft °F per ft).

First determine whether the temperature will be readily distributed within the bar, by computing the relative surface-internal resistance criterion hr/k. This is $6 \times (1/8)/26$ or 0.029, which is well below the maximum allowable value of 0.2 given above.

The surface area, ends plus cylindrical sides, is 0.883 sq ft. The volume is 0.049 cu ft. Equation 9.20 then gives

$$\frac{t\text{–}600}{70\text{–}600} = e^{-\frac{6 \times .883 \times (40/60)}{0.12 \times 450 \times 0.049}}$$

$$= e^{-1.33} = 0.262$$

$$t = 600 + 0.26(70 - 600)$$

$$= 461°\text{F}$$

Where the surface conductance is high with respect to the thermal conductivity of an object to be heated or cooled, the surface temperature changes faster than the interior, and equation 9.19b does not apply. It is necessary to consider local changes in temperature with time, which result from differences in temperature gradients. For transient heat flow, in one direction, an instantaneous heat balance gives

$$c\gamma \, dx \, dy \, dz \frac{\partial t}{\partial \theta} = k \, dy \, dz \frac{(\partial t/\partial x)}{\partial x} dx \tag{9.20}$$

Where thermal properties are constant with time, temperature, and position, this becomes

$$\partial t/\partial \theta = (k/c\gamma)(\partial^2 t/\partial x^2) \tag{9.21}$$

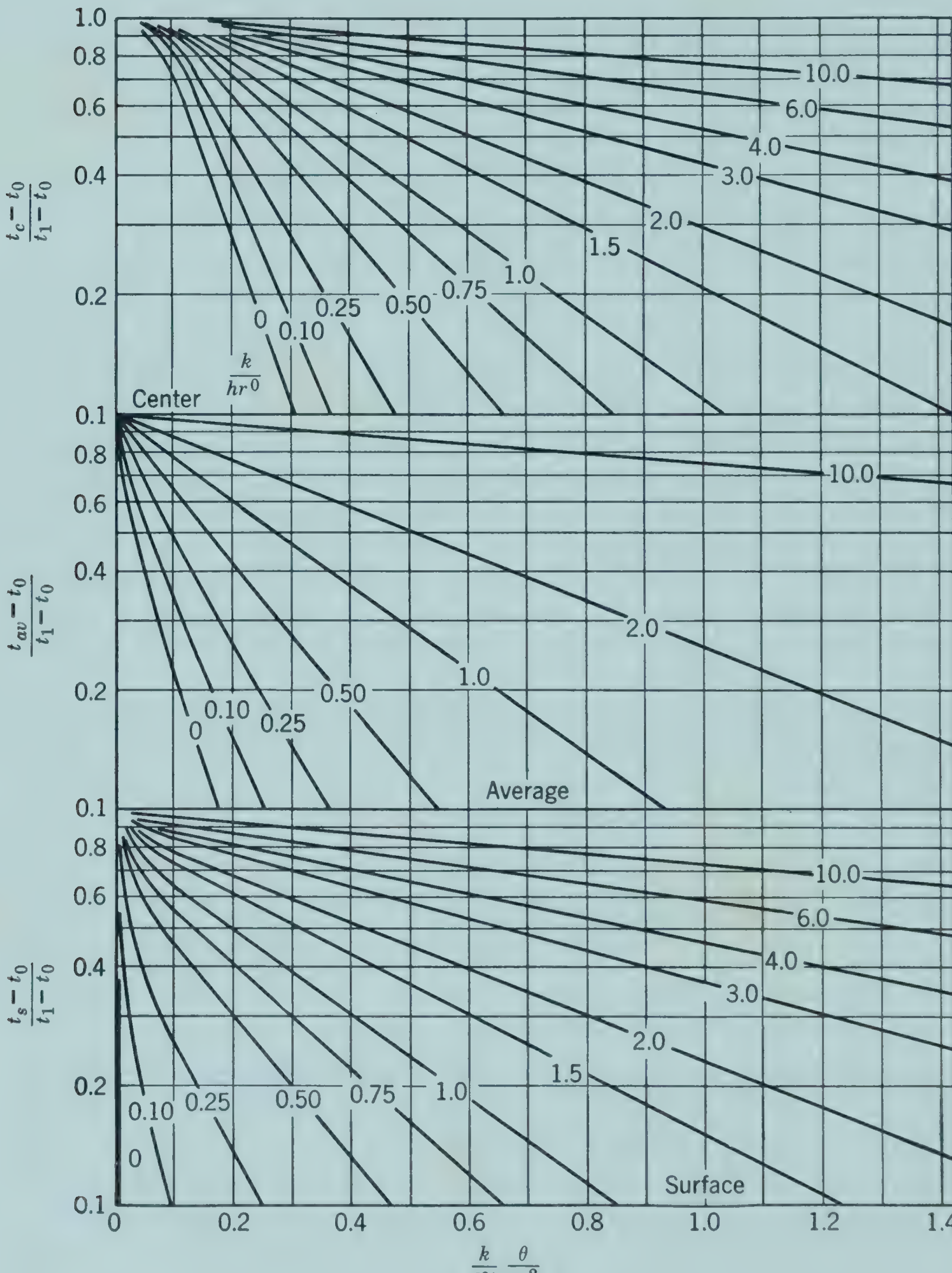


Fig. 9.2. Transient temperatures in a sphere.

which is known as Fourier's law of heat conduction in one dimension. Analytic solution of this equation is beyond the scope of

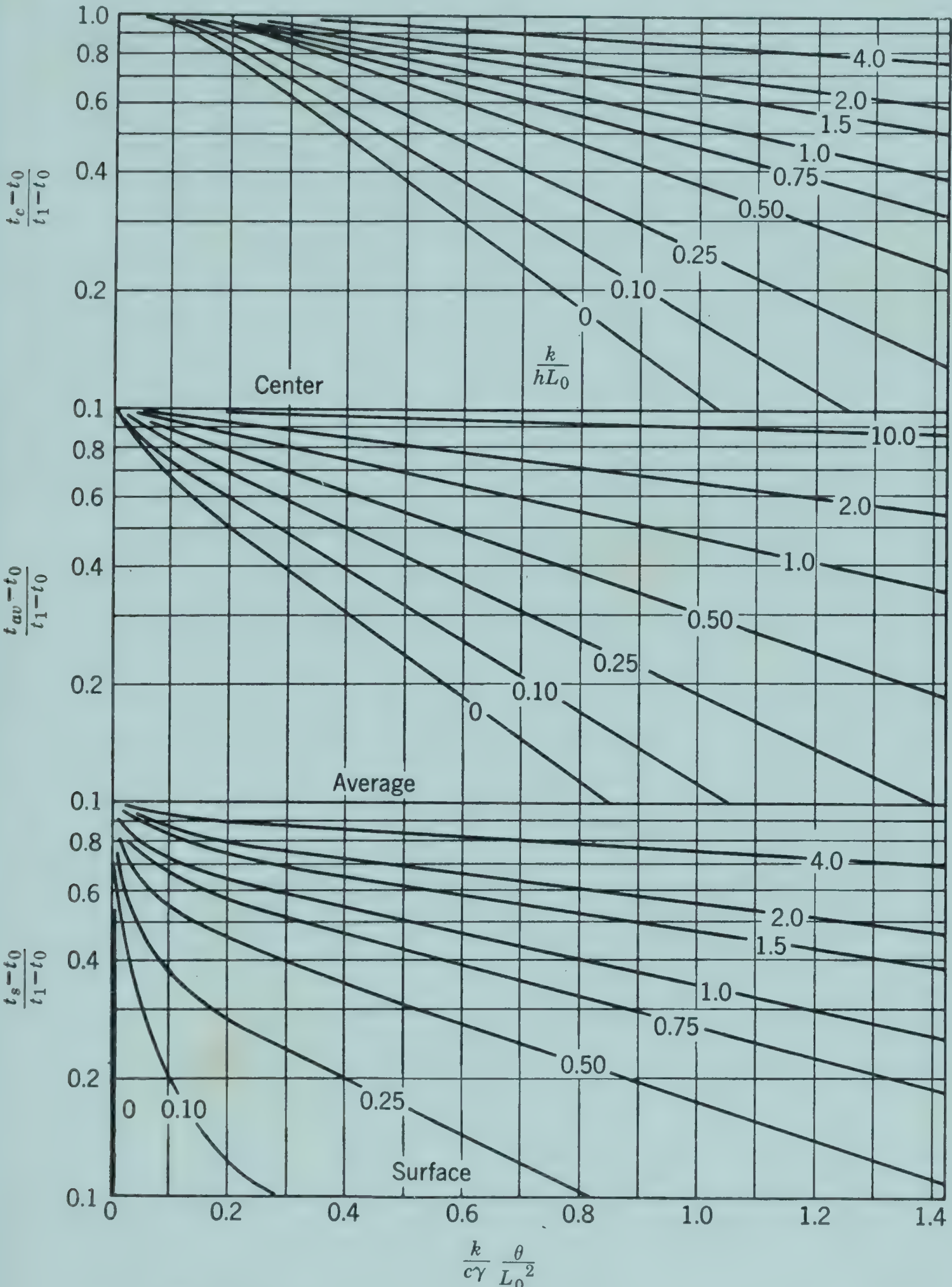


Fig. 9.3. Transient temperatures in a cylinder.

this book. It is presented here primarily to introduce the group of thermal properties $k/c\gamma$, known as the thermal diffusivity, which is significant in transient heat conduction. It is the ratio of the thermal conductivity to volumetric heat capacity. If two

objects of the same size and same thermal diffusivity are placed in the same surroundings, they will experience the same variations

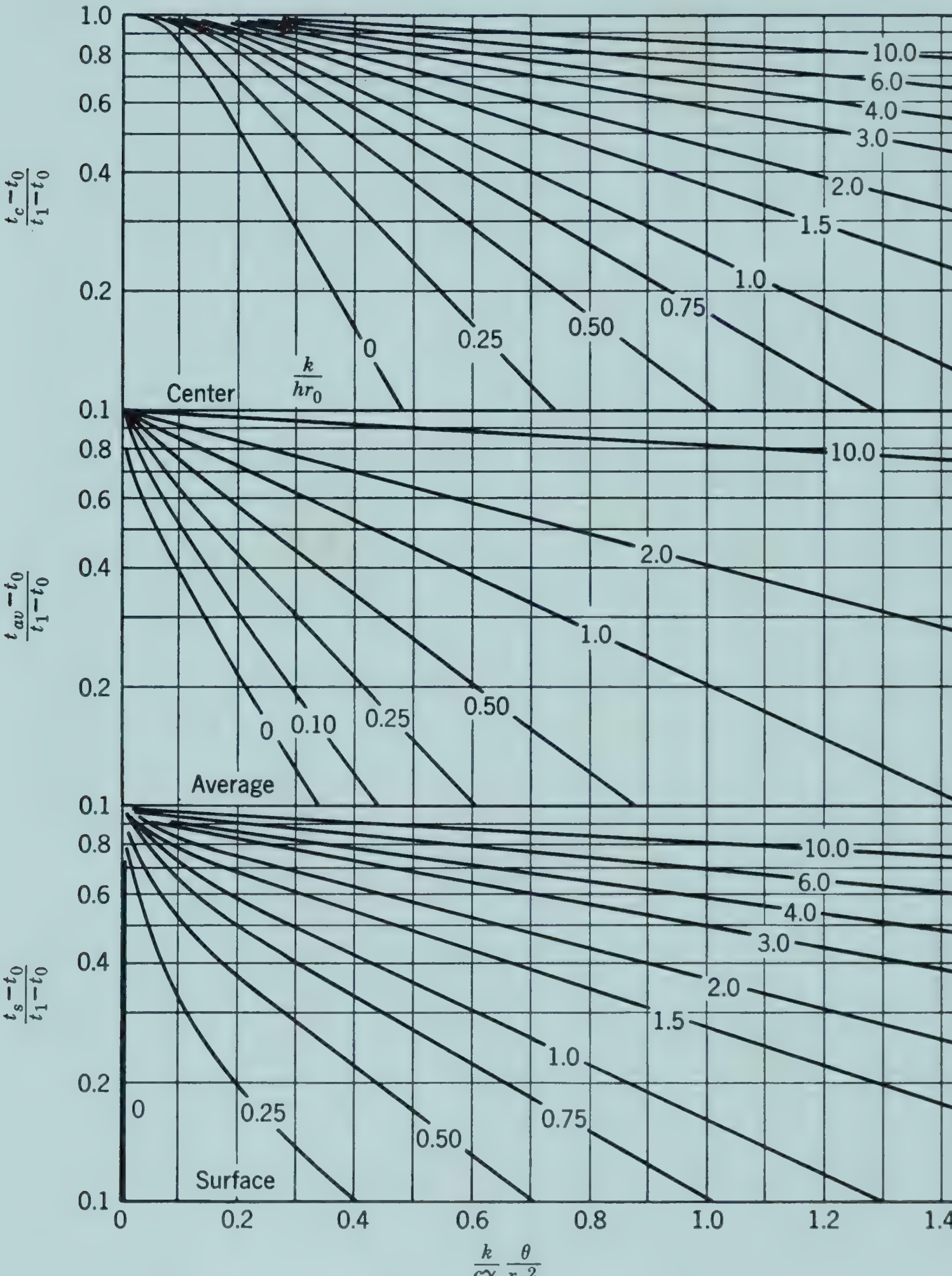


Fig. 9.4. Transient temperatures in a slab.

in temperature. The thermal diffusivity thus appears in one of the dimensionless moduli for transient heat flow, the Fourier number, or $(k/c\gamma)(\theta/r^2)$.

The temperature changes in a number of regular solids suddenly subjected to a change in temperature of surroundings have been found by several investigators, by analytic solution of equation 9.21, or from its extension to three dimensions. The resulting solutions are sums of infinite series, which can be presented in terms of four dimensionless moduli.

$$\frac{t - t_0}{t_1 - t_0} = \psi \left(\frac{k\theta}{c\gamma {r_0}^2}\right) \quad \left(\frac{hr_0}{k}\right) \quad \left(\frac{r}{r_0}\right) \tag{9.22}$$

Solutions [1,12,14] for the sphere, the infinitely long cylinder (or the short cylinder with perfectly insulated ends), and the slab (or the rectangular solid with insulated edges) are given in Figs. 9.2, 9.3, and 9.4.

θ = time, hr.
t_0 = temperature of surroundings, °F.
t_1 = initial temperature of solid, when $\theta = 0$, °F.
t_c, t_{av}, t_s = respective temperatures after time θ °F.
L_0 = one half slab thickness, ft.
r_0 = cylinder and sphere radius, ft.
ψ = a function of.

The temperature ratio for the short cylinder has been shown to be the product of the ratio for the infinitely long cylinder multi-

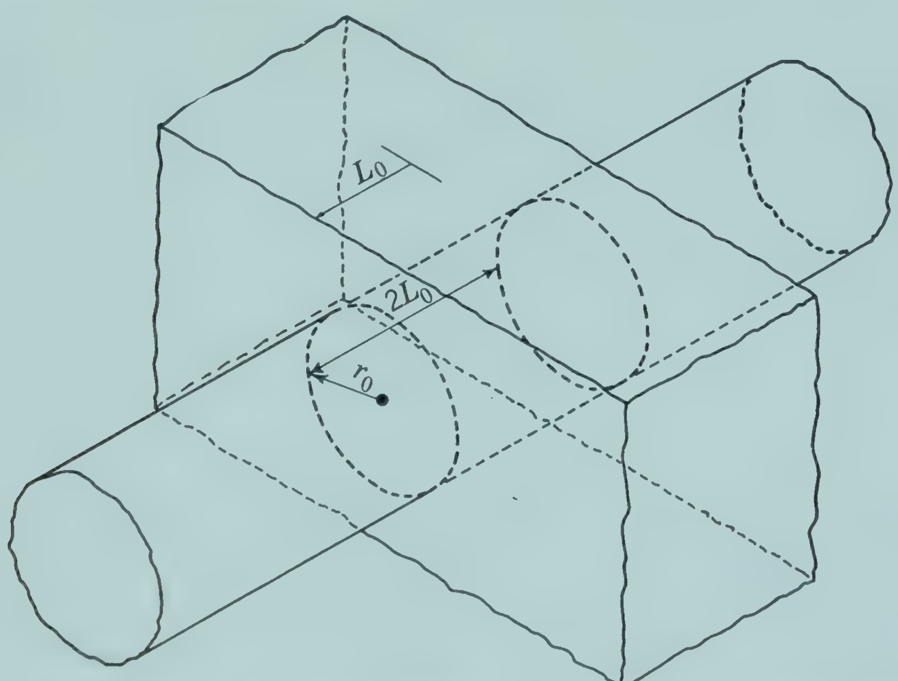


Fig. 9.5. For transient analysis, the short or finite cylinder is represented as the volume defined by the intersection of an infinitely long cylinder and a slab.

plied by the ratio for the infinite slab of a thickness that defines the length of the short cylinder, Fig. 9.5. Similarly, the ratio for a rectangular solid with all surfaces exposed is the product of the ratios for three slabs of thicknesses equal respectively to the three dimensions of the solid.

Example. A number 2 can, net diameter 3⅝16 in., net length 4¾16 in., contains 1.25 lb of a solid-pack food product which has a moisture content of 80 per cent and a specific weight of 68 lb per cu ft. The can, with its contents initially at 180°F, is suddenly placed in a retort where the surrounding temperature is promptly raised to 240°F with steam. Find the temperature at the center of the can at the end of 30 min. Surface conductance h for condensing steam, 1000 Btu per hr-ft^2 °F.

The specific heat is estimated to be 0.84 Btu per lb °F; the thermal conductivity, 0.25 Btu per (hr sq ft °F per ft). The thermal diffusivity is then $0.25/(0.84 \times 68)$ or 0.00438 sq ft per hr. The radius is $(3\frac{5}{16})/24$ or 0.138 ft. The half thickness of the plane slab which defines the length is $(4\frac{3}{16})/24$ or 0.174 ft.

For the cylinder

$$k/hr_0 = 0.25/(1000 \times 0.138) = 0.0018$$

At 30 min

$$k\theta/c\gamma r_0^2 = (0.00438 \times 0.5)/0.138^2 = 0.115$$

From Fig. 9.3, the residual temperature ratio is 0.8 at the center.

For the slab

$$k/hL_0 = 0.25/(1000 \times 0.174) = 0.0014$$

At 30 min

$$k\theta/c\gamma L_0^2 = (0.00438 \times 0.5)/0.174^2 = 0.0725$$

From Fig. 9.4, the residual temperature ratio is 0.95 at the center.

For the short cylinder, the residual temperature ratio is the product of the ratio for the long cylinder and the slab which defines the ends.

$$\frac{t - t_0}{t_1 - t_0} = 0.8 \times 0.95 = 0.76 \qquad \frac{t - 240}{180 - 240} = 0.76$$

From this

$$t = 240 + (180 - 240) \times 0.76$$
$$= 240 - 45.6 = 194.4°\text{F}$$

Analytic solutions are not readily available for many problems that involve irregular initial temperature distribution or in which boundary conditions are not constant or where properties change with time, temperature, or position. In some such cases, numerical methods [5,6,17] of solution can be used, or devices for solving analogous electrical problems may be available.

CONVECTION

Two principal kinds of problems arise in convection. As indicated by equation 9.2, the surface thermal conductance h_c and the temperature difference must both be established. This section will treat the estimation of conductances and a later section on heat exchangers will deal with temperature differences.

Much of the resistance to heat transfer by convection is found in the layer of fluid, in laminar flow without mixing, moving adjacent to the surface. Heat is transferred through this layer only by conduction. The surface conductance can be thought of as the conductance of a fictitious layer x_f, having the conductivity of the fluid, defined by $h_c = k/x_f$. The conductance can be increased by reducing the thickness of the laminar layer by more vigorous agitation, more active thermal circulation, or by operation at higher Reynolds-number values.

Although the concept of the fictitious equivalent layer is helpful in picturing convection resistance, it is not necessary to determine the thickness in convection calculations. A more direct approach is to establish conditions that give similar patterns of fluid flow, for which similar temperature patterns result. As would be expected, the Reynolds number Re ($DV\gamma/\mu$ or DG/μ) which is a criterion for similarity of fluid flow is a valid criterion in forced convection transfer. For natural or free convection transfer, a criterion that relates the buoyancy forces tending to promote fluid transport to the viscous and inertia flow resistances is required. Such a criterion ($D^3\gamma^2 g\beta\,\Delta t/\mu^2$) has been named the Grashof number, Gr.

For convection, a modulus that will include the thermal conductance is obviously required. The criterion (hD/k), named Nu for Nusselt, has been found valid. Since this can be written as $D/(k/h_c) = D/x_f$, it can be thought of as the ratio of a pertinent significant physical dimension, such as the diameter of a pipe, to the fictitious layer thickness. The influence of the thermal properties is included by the modulus ($c\mu/k$) which is designated the Prandtl number, Pr. In systems where the flow pattern changes from point to point, as it does near the entrance of a pipe, a relative-position modulus such as (L/D) is required.

For free convection, the general expression is then

$$\frac{hD}{k} = \psi_n \left(\frac{D^3\gamma^2 g\beta\, \Delta t}{\mu^2}\right)\left(\frac{c\mu}{k}\right)\left(\frac{L}{D}\right) \tag{9.23}$$

For forced convection, the general expression is

$$\frac{hD}{k} = \psi_f \left(\frac{DV\gamma}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{L}{D}\right) \tag{9.24}$$

The functions ψ_n and ψ_f for natural and for forced convection have been determined experimentally for many systems in commercial and industrial use. In some cases where they are not simple functions, approximate power functions are used over specified ranges of the variables.

9.8. Free Convection. For free convection about horizontal cylinders, equation 9.23 can be represented by a power function.

$$\frac{hD}{k} = C\left[\frac{D^3\gamma^2 g\beta\, \Delta t}{\mu^2}\,\frac{c\mu}{k}\right]^n \tag{9.25}$$

Over the range of $10^4 < Gr \cdot Pr < 10^9$, $C = 0.53$, $n = 0.25$. Above $Gr \cdot Pr$ of 10^9, $C = 0.12$ and $n = \frac{1}{3}$.

9.9. Free Convection, Gases. For gases it is convenient to solve equation 9.25 for h and then to group the constants and thermal properties as follows, for the lower range,

$$h = 0.53\left[\frac{\gamma^2 g\beta c k^3}{\mu}\right]^{0.25}\left(\frac{\Delta t}{D}\right)^{0.25} = b\left(\frac{\Delta t}{D}\right)^{0.25} \tag{9.26}$$

The coefficient b is a particular function of temperature for each gas and includes the effect of temperature upon the thermal properties. For air, $b = 0.288(1 - 0.00057t)$. Since b varies only slightly with air temperature at normal temperatures, it is convenient to use a mean value of 0.27. For air, the equation then becomes

$$h = 0.27\,(\Delta t/D)^{0.25} \tag{9.27}$$

For shapes other than horizontal cylinders, equations similar to equation 9.25 are available, but with constants appropriate for the particular shape, a steam radiator, for example, or a bank of tubes.

For plane surfaces in air:

1. Horizontal, heated facing upward, or cooled facing downward, over 3 ft square,

$$h = 0.38\ (\Delta t)^{0.25} \tag{9.28}$$

2. Horizontal, heated facing downward, or cooled facing upward, over 3 ft square,

$$h = 0.2\ (\Delta t)^{0.25} \tag{9.29}$$

3. Vertical surfaces over 1 ft high

$$h = 0.27\ (\Delta t)^{0.25} \tag{9.30}$$

4. Vertical air spaces, over 1 ft high and more than 1 in. wide. Equation 9.30 can be applied, on recognizing that two resistances occur in series, with half the over-all surface-to-surface difference available for each. Thus

$$h = \frac{0.27}{2}\left(\frac{\Delta t}{2}\right)^{0.25} = 0.12\ (\Delta t)^{0.25} \tag{9.31}$$

For vertical spaces less than 1½ in. wide, convection is restricted, being almost suppressed in spaces less than ¼ in. wide. Conductances for air spaces from ⅛ to 1½ in. wide, reported by Rowley and Algren,[8] include the radiation which occurs in parallel with the convection.

Example. Find the rate of heat loss per lineal foot from a 2-in. bare horizontal steampipe, with a surface temperature of 330°F, in a room at 70°F. The outside diameter of 2 in. nominal pipe is 2.38 in.

The temperature difference is 330 − 70 or 260°F. The outside diameter is 0.198 ft. Equation 9.27 then gives

$$h = 0.27(260/0.198)^{0.25} = 1.62 \text{ Btu per (hr sq ft °F)}$$

The loss rate per lineal foot, by equation 9.2 (for convection only),

$$q = 1.62 \times 0.198 \times \pi(330 - 70) = 263 \text{ Btu per hr ft}$$

9.10. Free Convection, Liquids. For liquids the coefficients of thermal expansion and viscosity change markedly with temperature. An expression similar to equation 9.27 is of doubtful utility because the limits within which it is applicable are not readily recognized. Instead, equation 9.25 is rewritten

$$hD/k = C(aD^3\ \Delta t)^n \tag{9.32}$$

in which it can be seen that

$$a = \gamma^2 g\beta c/\mu k \tag{9.33}$$

Tabular values of a for particular temperatures for a given fluid can be computed. From these, an approximate algebraic representation may be developed. For water, between 50° and 180°F,

$$a = 6.3 \times 10^4(t_f - 10)^2 \tag{9.33a}$$

in which t_f is the mean temperature of the laminar layer, midway between the surface and bulk fluid temperatures. The constants and exponents in this expression have no special physical significance but are simply a convenient way of summarizing data on water.

Since water has a maximum density at 39°F, equations 9.32 and 9.33 cannot be used for temperature differences which span 39°F. Where it is desired to cool quantities of water with ice, temperatures below 39°F are not readily secured with ice floating in a tank. As water adjacent to the ice begins to cool below 39°F, it tends to remain at the top of the tank, with very poor convection transfer of its heat to the ice. A more effective arrangement is a spray or shower of recirculated water over cakes of ice supported above the water level on a screen rack.

Example. Find the heat-transfer coefficient from a 3-in. diameter steampipe, with a surface of 210°F, to water in a water-heater tank at 150°F. The outside diameter of 3-in. nominal size pipe is 3.5 in. or 0.292 ft.

Equation 9.33 can be used, with values for water from equation 9.33*a*. The mean film temperature is $(210 + 150)/2$ or 180°F. From equation 9.33*a*, $a = 6.3 \times 10^4(180 - 10)^2$ or 18.2×10^8. In equation 9.25, the group in the bracket, which is the Grashof-Prandtl number product is $18.2 \times 10^8 \times 0.292^3(210 - 150)$ or 2.72×10^9. Since this is larger than 10^9, the value of C (equation 9.25) is 0.12 and n is ⅓. Thus

$$hD/k = 0.12(2.72 \times 10^9)^{1/3} = 0.12 \times 1395 = 167$$

and, since k for water = 0.389 at 180°F,

$$h = 0.389 \times 167/0.292 \text{ or } 223 \text{ Btu per (hr sq ft °F)}$$

9.11. Convection Transfer in Boiling. When heating surfaces are at temperatures above the boiling points of the liquids in which they are submerged, much greater heat transfer will occur than predicted from equation 9.25, because of the violent

agitation resulting from surface boiling. The conductances vary greatly with the arrangement, nature, and condition of the surface. Very roughly, for water, for Δt less than 40

$$h = 100\ (\Delta t) \tag{9.34}$$

At temperature differences greater than 40°F, vapor formation becomes so rapid that vapor blanketing of the surface occurs, and greater differences give no greater transfer. There may be danger of severely overheating the surface of direct-fired vessels. For organic liquids, the boiling conductance may be only $\frac{1}{10}$ to $\frac{1}{2}$ that for water.

9.12. Surface Condensation of Vapors. In transfer of heat from condensing vapors to condenser surfaces, the principal resistance exists in the film of condensed liquid adhering and draining slowly from the surface. The recommended value for condensing steam is $h = 1000$. With particular care to obtain dropwise condensation instead of film condensation, values as large as 6000 have been secured. For organic vapors, a conservative value is 200.

Air or other noncondensable gas which may enter with the vapor will tend to be concentrated at the vapor-condensate interface, and offer substantial resistance to condensation. Steam jet ejectors or dry-vacuum pumps are used to remove noncondensable gas from surface or jet condensers. Noncondensable gas purgers are installed on large ammonia refrigeration systems.

FORCED CONVECTION

Forced convection is employed in two general types of systems, (*a*) fluids flowing in pipes of circular or annular cross section, and (*b*) fluids flowing across single pipes, or finned tubes, and other flat or irregular objects. The general Nusselt equation, 9.24, can, with appropriate constants, be adapted to each.

9.13. Forced Convection Inside Pipes. For heat transfer in long pipes, a single power function has been found adequate. Since turbulence is well developed by the entrance conditions in much industrial apparatus and only mean values over the length of the pipe are desired, the (L/D) group is usually omitted.

$$hD/k = 0.023(DG/\mu)^{0.8}(c\mu/k)^{0.4} \tag{9.35}$$

is recommended by McAdams [12] for fluids of viscosities not more than twice that of water. The Reynolds number must be above 2100 to insure turbulent flow. In equation 9.35, the thermal properties are to be evaluated at the bulk mean temperature. With liquids that have a significant change in viscosity with temperature, the laminar layer will be warmer and less viscous than the bulk of the fluid in heating, and colder and more viscous in cooling. Dittus and Boelter [12] recognized this by using a constant of 0.0243 and an exponent of 0.4 for the Prandtl number for heating, and a constant of 0.0265 with a Prandtl exponent of 0.3 for cooling. However, McAdams [12] prefers instead the single equation of Sieder and Tate [12] for fluids of high viscosity

$$\frac{hD}{k} = 0.027 \left(\frac{\mu}{\mu_s}\right)^{0.14} \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \tag{9.36}$$

In equation 9.36, the fluid properties are evaluated at the bulk mean temperature, except μ_s which is found at the temperature of the surface.

For a particular fluid, application of equation 9.35 or 9.36 can be simplified by solving for h, with the thermal properties and constants grouped together and expressed as a function of temperature. For design calculations, it is desirable to give diameter in inches, $D = D'/12$, and to use the flow rate in pounds per hour per tube, $w = G\pi(D')^2/576$, instead of the unit mass velocity G. Equation 9.35 is rewritten as

$$h = \left[0.023 \left(\frac{576}{\pi}\right)^{0.8} 12^{0.2} \frac{c^{0.4}k^{0.6}}{\mu^{0.4}}\right] \frac{w^{0.8}}{(D')^{1.8}} \tag{9.37}$$

$$= fw^{0.8}/(D')^{1.8} \tag{9.38}$$

in which f, defined by the constants and properties in the bracket, is a function of temperature. When values of f for several representative temperatures are plotted against t, a very nearly linear relation is found. For water, between 40 and 210°F

$$f = 0.53(1 + 0.01t)$$

For pipes of noncircular cross section, equation 9.35 or 9.36 can be employed, using for D the diameter equivalent to the hydraulic radius, $D_e = 4r_{\text{hydr.}}$ With annular spaces, $D_e = D_2 - D_1$.

If flow is in the streamline region (Reynolds number less than 2100), no lateral mixing occurs, and heat must be transferred entirely by conduction through laminar layers. Treatment of this region and the transition region (up to Reynolds number of 4000) is somewhat more involved than for the turbulent flow and is beyond the scope of this text. It is treated in detail by McAdams.[12] Since heat-transfer coefficients are poor in these regions, design and operating conditions should be arranged to give turbulent flow.

Example. Estimate the surface thermal conductance for heating "rough-broken" tomato pulp from 70° to 170°F, at the rate of 30,000 lb per hr in a tube 2.33 in. in inside diameter. The tube surface temperature will be 220°F. Thermal properties are:

Density, 64 lb per cu ft.
Solids content, 6 per cent.
Estimated specific heat, 0.95 Btu per lb °F.
Viscosity, 3.5 centipoise at 68°F.
1.9 centipoise at 135°F.
1.3 centipoise at 200°F.

Thermal conductivity, estimated at 95 per cent of that of water. The bulk mean temperature is $(70 + 170)/2$ or 120°F. At this temperature, the viscosity, interpolated from the data, is 2.15 centipoises, which is equal to 2.42×2.15 or 5.2 lb per (hr ft). The thermal conductivity, at 120, is 0.95×0.368 or 0.35 Btu per (hr sq ft °F per ft). Pipe cross section is $0.194^2 \times 0.7854$, or 0.0296 sq ft.

Since the viscosity is more than twice that of water, equation 9.36 will be used. The mass velocity G is $30{,}000/0.0296 = 1.013 \times 10^6$ lb per (hr sq ft). The Reynolds number is $0.194 \times 1.013 \times 10^6/5.2 = 37{,}800$, which is well into the turbulent zone. The Prandtl number is $0.95 \times 5.2/0.35 = 14.1$. The viscosity at the surface temperature of 220°F, estimated by extrapolation, is 1.2 centipoises, or $2.42 \times 1.2 = 2.9$ lb per (hr ft). Inserting these values in equation 9.36 gives

$$\frac{hD}{k} = 0.027 \left(\frac{5.2}{2.9}\right)^{0.14} (37{,}800)^{0.8}(14.1)^{1/3}$$

$$= 324$$

From this,

$$h = 324 \times 0.35/0.194 = 585 \text{ Btu per (hr sq ft °F)}$$

Before proceeding with a full-scale design, it would be advisable to ascertain by laboratory tests at the same Reynolds number and tube-surface temperature, whether any significant reduction in conductance will occur because of accumulation of "cooked" film on tube surfaces.

9.14. Forced Convection Across Pipes, Banks of Pipes, and Plates. For forced convection across single cylinders and banks of tubes, equation 9.24 cannot be represented by a single power function for the whole range of Reynolds numbers for which data are available. It may be well to remark here that the Reynolds number for flow across cylinders, being based on the outside diameter of the cylinder and the mass velocity in the free cross section, has no direct relation to the Reynolds number for flow in pipes. Laminar flow may prevail from the leading edge to the diameter normal to the flow, and a turbulent wake occurs even at low Reynolds numbers. The numerical value of 2100, which is the lower limit for turbulent flow in pipes, has no particular significance here.

9.15. Forced Convection Across Single Cylinders. For gases flowing normal to single cylinders,[12] in the range of Reynolds numbers from 0.1 to 1000

$$\frac{hD_0}{k}\left(\frac{k}{c\mu}\right)^{0.3} = 0.35 + 0.47\left(\frac{D_0G}{\mu}\right)^{0.52} \tag{9.39}$$

While from Reynolds numbers of 1000 to 50,000

$$\frac{hD_0}{k}\left(\frac{k}{c\mu}\right)^{0.3} = 0.26\left(\frac{D_0G}{\mu}\right)^{0.6} \tag{9.40}$$

For liquids flowing normal to single cylinders, in the range of Reynolds numbers from 0.1 to 200, the curve has the form of equation 9.40, but with a constant of 0.86 and an exponent of 0.43. Above Reynolds numbers of 200, equation 9.39 is recommended. In each of these equations, the fluid properties are to be evaluated at the film temperature, the arithmetic average of surface and bulk mean fluid temperatures.

For spheres, in the range of Reynolds numbers from 20 to 150,000, the equation is similar to equation 9.40, but the constant is 0.36.

9.16. Forced Convection Across Banks of Tubes. Banks of tubes can be represented by equations similar to equation 9.40. The mass velocity G, to be used is the rate of flow, divided by the minimum free area, whether it occurs in transverse or diagonal spacing. In a bank of in-line tubes, the second row has been shown to have a lower conductance than the first and the subse-

quent rows. In a bank of staggered tubes, the second and third tubes may be from 3 to 20 per cent higher than the first, depending upon the spacing. However, the pressure drop for staggered tubes is higher than that for in-line tubes.

With factory-assembled finned tubes, particular constants for an assembly are determined by test. Expressions such as equation 9.40 aid in correlating the data.

9.17. Forced Convection, Plane Surfaces. Where a fluid flows parallel to a plane surface unconfined by the walls of a duct, the flow pattern is at first considerably influenced by the nature of the leading edge. The distance from the leading edge is the significant dimension to be used in the Nusselt and Reynolds numbers. A laminar layer develops at the leading edge if it is gently rounded. A turbulent zone may start at a blunt leading edge, or it may not develop till Reynolds numbers of 100,000 to 500,000 are reached following a gently rounded edge. The surface conductance is high where the laminar layer is initially thin, drops as the layer thickens, rises in the transition zone, and then gradually drops. For mean values of the thermal conductance, Jakob [10] gives an equation similar to equation 9.35, but with a constant of 0.031, for Reynolds numbers above 200,000.

Very careful measurements were made by Rowley, et al.,[8] of surface thermal conductances for air flowing in a 6-by-12-in. duct parallel to surfaces of several typical building materials. The results are expressed simply as

$$h_{c+r} = 1.5 + 0.164V \quad \text{for glass and smooth paint} \tag{9.41}$$

$$h_{c+r} = 1.8 + 0.168V \quad \text{for smooth plaster} \tag{9.42}$$

$$h_{c+r} = 2 + 0.218V \quad \text{for concrete} \tag{9.43}$$

$$h_{c+r} = 2 + 0.31\ V \quad \text{for stucco} \tag{9.44}$$

In these equations, the subscript of h denotes that it is for combined convection and radiation. V is in feet per second. The values given are for a mean temperature (arithmetic mean of surface and its exposed surroundings) of 20°F, representative of outdoor winter conditions in cold climates, at which the radiation conductance is 0.7. Subtracting 0.7 gives convection conductances. Since the net values for zero velocity must be for free convection (at about 45 degrees temperature difference) the

curves are essentially tangents drawn from the free-convection intercepts to $(V)^{0.8}$ curves to represent forced convection. Appropriate tangents can be drawn for other free-convection conditions, and also pertinent radiation conductances can be added for other radiant conditions.

RADIATION

9.18. Emissivity. Radiant energy from a perfect emitter, known technically as a black body, is emitted in a continuous spectrum of wave lengths according to Planck's law

$$W_{b\lambda} = \frac{1.16 \times 10^8 \lambda^{-5}}{e^{(25740/\lambda T)} - 1} \tag{9.45}$$

in which $W_{b\lambda}$ = the monochromatic or spectral emissive power of a black body, in Btu per (hr sq ft micron). One micron is $\frac{1}{1000}$ mm.
λ = the wave length in microns.
e = the Naperian base, 2.7183.
T = the absolute temperature, degrees Rankine.

The integral of $W_{b\lambda}\,d\lambda$ from wave length 0 to infinity gives the black-body emissive power per square foot, $W_b = \sigma T^4$, or $0.173 \times 10^{-8} T^4$ which was included in equation 9.3.

Monochromatic emissive power for a black body at several temperatures is shown in Fig. 9.6. The maximum value of $W_{b\lambda}$ of $217.5 \times 10^{-15} T^5$ at a given temperature T occurs at a wave length of $\lambda_{max} = 5193/T$.

Monochromatic emissivity ϵ_λ is defined as the ratio of the emissive power of a nonblack radiator in a given wave length to black-body monochromatic emissive power in the same wave length. For most surfaces, it varies with wave length, as shown in Fig. 9.7. A surface having a constant monochromatic emissivity is called a gray body.

Mean emissivity values

$$\epsilon = \frac{\int_0^\infty \epsilon_\lambda W_{b\lambda}\,d\lambda}{W_b} \tag{9.46}$$

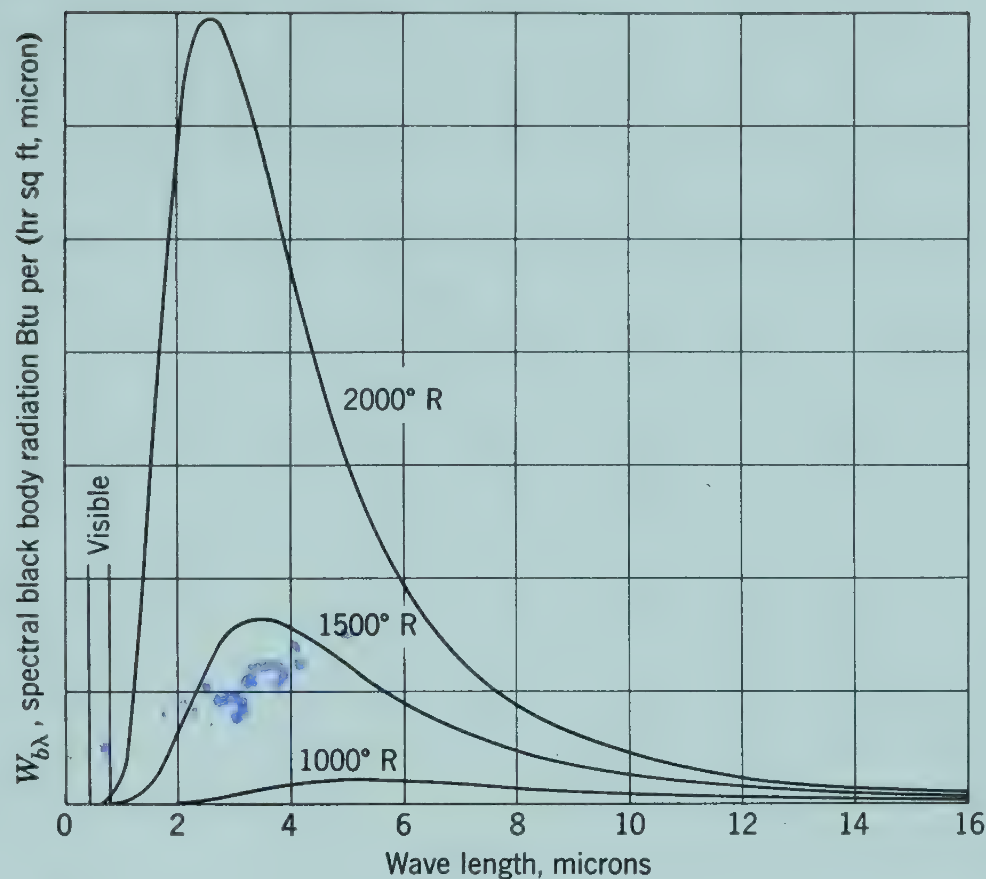


Fig. 9.6. Monochromatic emissive power of a black body.

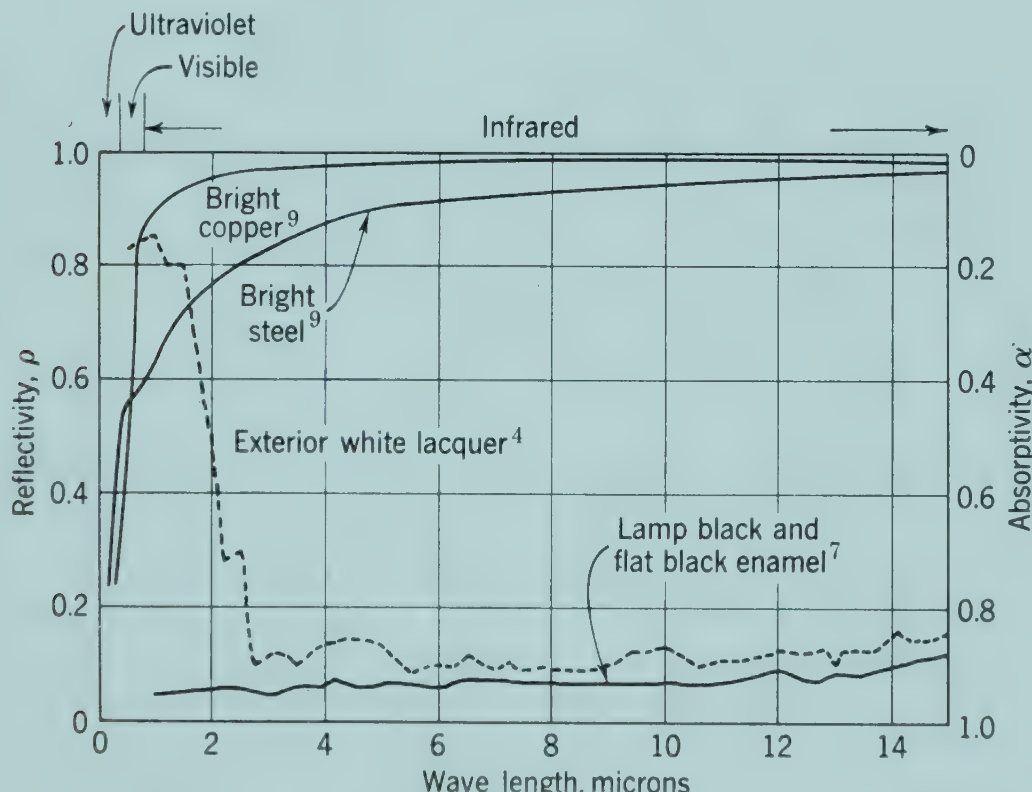


Fig. 9.7. Monochromatic emissivity of four surfaces.

can be measured directly in the laboratory (with proper precautions to minimize and correct for convection and counterradiation) at normal temperatures and up to as high temperatures as the material can endure. Since mean values depend upon temperature as well as monochromatic emissivity, the temperature should be stated when a value of emissivity is given, as in Table 9.3. Emissivities do not change appreciably with temperature below 600°F.

Table 9.3 NORMAL TOTAL EMISSIVITY OF VARIOUS SURFACES *

Surface	*Temperature, °F*	*Emissivity*
Aluminum, polished plate	73	0.040
Aluminum, oxidized at 1110°F	390–1110	0.11–0.19
Copper, polished	242	0.023
Copper, heated to 1110°F	390–1110	0.57
Polished iron	800–1880	0.144–0.377
Ground sheet steel	1720–2010	0.55–0.61
Oxidized iron	212	0.736
Steel plate, rough	100–700	0.94–0.97
Nickel, 98.9% pure, polished	440–710	0.07–0.087
Nickel plate, heated to 1110°F	390–1110	0.37–0.48
Zinc, 99.1% pure, polished	440–620	0.045–0.053
Galvanized sheet iron, fairly bright	82	0.228
Galvanized sheet iron, gray oxidized	75	0.276
Asbestos, paper	100–700	0.93–0.945
Enamel, white fused on iron	66	0.897
Glass, smooth	72	0.937
Oak, planed	70	0.895
Snow-white enamel varnish on rough iron plate	73	0.906
Flat black lacquer	100–200	0.96–0.98
Oil paints, 16 different, all colors	212	0.92–0.96
Aluminum paint, 10% Al, 22% lacquer body	212	0.52
Aluminum paint, 26% Al, 27% lacquer body	212	0.3
Aluminum paints, varying age and Al content	212	0.27–0.67
Paper	66	0.924–0.944
Porcelain, glazed	72	0.924
Roofing paper	69	0.91
Water	32–212	0.95–0.963

* Adapted from McAdams.[12]

9.19. Absorptivity. In the transfer of radiant energy, absorption is as important as emission. Bodies that emit radiant energy can also absorb it. The radiant energy absorbed is converted into heat. Roughly, bodies that are good emitters are also good absorbers. At normal temperatures, a surface that is a good absorber in the visible region of the spectrum appears black to the eye because it reflects no radiation, and, being at normal temperature, it also emits none in the visible region. A significant amount of energy in the visible region is emitted at high temperatures. For example, at 2000°F, a black line on a china plate appears brighter than the white surface because, being a better absorber than the white in the visible region, it is in this region a better emitter.

Monochromatic absorptivities α_λ are usually measured indirectly by measuring reflectivities ρ_λ. For opaque objects, impinging energy is either absorbed or reflected; therefore $\alpha = 1 - \rho$.

It can be proved that

$$\alpha_\lambda = \epsilon_\lambda \tag{9.47}$$

This can be generalized into $\alpha = \epsilon$ for gray and black bodies; but for selective absorbers the generalization applies only when they are absorbing at the same temperature at which they are emitting. Mean absorptivity depends not only upon monochromatic absorptivity but also on the spectral distribution of the impinging energy,

$$\alpha = \frac{\int_0^\infty \alpha_\lambda \Gamma_\lambda d\lambda}{\Gamma} \tag{9.48}$$

in which Γ_λ is the monochromatic irradiation, Btu per (hr sq ft micron).

Γ is the total irradiation, Btu per (hr sq ft).

Example. Find the absorptivity for white paint shown in Fig. 9.7, for black-body radiation from a surface at 540°F (1000°R).

From Fig. 9.6, black-body radiation at 1000°R is seen to be confined to the region from 3 to 15 microns. Since the white paint is a good reflector only in wave lengths less than 3 microns and has a relatively constant absorptivity of about 0.12 from 3 to 15 microns, its absorptivity in this range can be taken by inspection from the curve at 0.12. A more precise value

could be obtained by inserting point values into equation 9.48 and integrating numerically.

9.20. Radiant Heat Exchange. An object that is radiating is in turn subjected to irradiation from surrounding surfaces and absorbs part of this irradiation. Usually net radiant gain or loss is desired. For concentric spheres or infinite concentric cylinders, the net rate is

$$q = \frac{A_1\sigma(T_1{}^4 - T_2{}^4)}{\dfrac{1}{\epsilon_1} + \dfrac{A_1}{A_2}\left(\dfrac{1}{\epsilon_2} - 1\right)} \tag{9.49}$$

If A_1 is very small compared to A_2, equation 9.49 becomes the expression for a small object completely surrounded.

$$q = A_1\epsilon_1\sigma(T_1{}^4 - T_2{}^4) \tag{9.50}$$

Example. Find the net radiant-heat loss rate per lineal foot from a 2-in. bare steam pipe with a surface temperature of 330°F in surroundings at 70°F. How does the radiant loss rate compare with the convection loss rate as illustrated in the example in sect. 9.9?

This can be solved by the use of equation 9.50, for a small object completely surrounded by other surfaces that are at a uniform temperature. The emissivity is estimated from Table 9.3, for rough steel plate to be 0.95. The area per lineal foot is $2.38 \times 3.1416/12$ or 0.62 sq ft per lineal ft.

$$T_1 = 330 + 460 \qquad T_2 = 70 + 460$$

$$\begin{aligned} q &= 0.62 \times 0.95 \times 0.173 \times 10^{-8}(790^4 - 530^4) \\ &= 0.62 \times 0.95 \times 0.173 \times [(\tfrac{790}{100})^4 - (\tfrac{530}{100})^4] \\ &= 316 \text{ Btu per hr-lin-ft} \end{aligned}$$

This is seen to be $^{316}\!/_{263}$ or 1.19 times the convection loss. The radiation is equivalent to a surface conductance of

$$\frac{316}{0.62 \times (330 - 70)} \quad \text{or} \quad 1.96 \text{ Btu per (hr sq ft °F)}$$

As A_1 approaches A_2 in area, equation 9.49 becomes the expression for infinite parallel planes. The emissivities appear thus because of multiple reflections.

For exchange between small elements of surfaces, the fraction of the energy from one which is intercepted by the other must be sought. The energy W, Btu per (hr sq ft), radiated from an element of surface is emitted in all directions into the surrounding

hemisphere. The energy rate in a particular direction is needed. The energy rate, in Btu per (hr sq ft normal to the direction of a ray) per unit solid angle, ω is called the intensity I. For a diffuse radiator,

$$I = W/\pi \tag{9.51}$$

Consider radiant exchange between two elements dA_1 and dA_2 whose normals make the angles ϕ_1 and ϕ_2 respectively with the line r drawn between their centers. The energy rate from dA_1 to dA_2 is

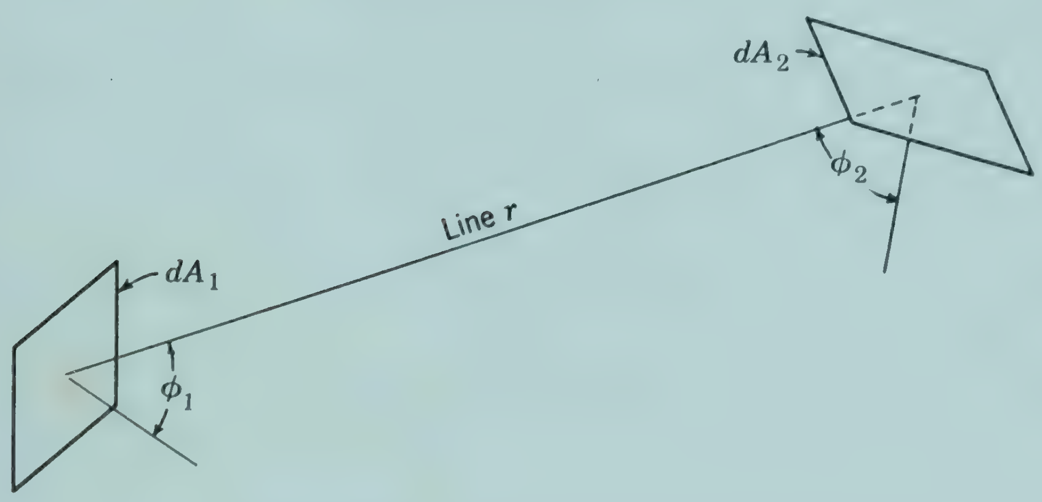
$$d^2q_{1\to 2} = I_1 \cos\phi_1 \, dA_1 \, d\omega_1$$

where

$$I_1 = W_1/\pi = \epsilon_1 \sigma T_1{}^4/\pi \tag{9.52}$$

and

$$d\omega_1 = \cos\phi_2 \, dA_2/r^2$$



Thus the energy from dA_1 which impinges on dA_2 is

$$d^2q_{1\to 2} = \frac{\epsilon_1 \sigma T_1{}^4}{\pi} \cos\phi_1 \, dA_1 \frac{\cos\phi_2 \, dA_2}{r^2} \tag{9.53}$$

Of this, the energy absorbed by dA_2 is $\alpha_{2,\,1} d^2q_{1\to 2}$. Similarly, the energy radiated from dA_2, absorbed by dA_1 is

$$\alpha_{1,\,2} d^2q_{2\to 1} = \alpha_{1,\,2} \frac{\epsilon_2 \sigma T_2{}^4}{\pi} \cos\phi_2 \, dA_2 \frac{\cos\phi_1 \, dA_1}{r^2} \tag{9.54}$$

So the net interchange, neglecting multiple reflections, is

$q_{1\to 2}$ net

$$= \iint (\alpha_{2,\,1}\epsilon_1 \sigma T_1{}^4 - \alpha_{1,\,2}\epsilon_2 \sigma T_2{}^4) \frac{\cos\phi_1 \cos\phi_2 \, dA_1 \, dA_2}{\pi r^2} \tag{9.55}$$

Define a shape factor F_1 as the fraction of the energy emitted from A_1 which falls on A_2; also define F_2 as the fraction from A_2 which falls on A_1. With these definitions equation 9.55 can be re-written as

$$q_{1\to 2},\ \text{net} = \alpha_{2,\,1}\epsilon_1\sigma T_1{}^4A_1F_1 - \alpha_{1,\,2}\epsilon_2\sigma T_2{}^4A_2F_2 \qquad (9.56)$$

From equation 9.56, it can be seen that if $T_2 = T_1$, then

$$A_1F_1 = A_2F_2 \qquad (9.56a)$$

Since these are simply geometric relationships, equation 9.56*a* must be generally true. Also, if the spectral nature of W_1 is about the same as W_2, then the absorptivities and emissivities are equal. Substitute in equation 9.56 the equivalent of A_2F_2 and also that of $\alpha_1 = \epsilon_1$ and $\alpha_2 = \epsilon_2$, to obtain

$$q_{1\to 2},\ \text{net} = \epsilon_1\epsilon_2A_1F_1\sigma(T_1{}^4 - T_2{}^4) \qquad (9.57)$$

From equations 9.55 and 9.56, it can be seen that

$$A_1F_1 = \iint \frac{\cos\phi_1 \cos\phi_2\, dA_1\, dA_2}{\pi r^2} \qquad (9.58)$$

$$A_2F_2 = \iint \frac{\cos\phi_1 \cos\phi_2\, dA_1\, dA_2}{\pi r^2} \qquad (9.59)$$

When A_1 and A_2 are small compared with r^2,

$$F_1 \cong \frac{\cos\phi_1 \cos\phi_2 A_2}{\pi r^2} \qquad (9.60)$$

$$F_2 \cong \frac{\cos\phi_1 \cos\phi_2 A_1}{\pi r^2} \qquad (9.61)$$

Example. Find the net radiant-energy rate from the inside top surface of an oven, 18 in. by 20 in. in size, at 500°F, and the top of a cake 9 in. square, which is at 160°F and is 8 in. below the oven top. The center of the pan is directly below the long axis of the oven and 4 in. in front of the center of the top.

This problem can be solved by equation 9.57, but first the shape factor, or fraction of radiant energy from the top which strikes the cake, must be

found. Use the approximation of equation 9.60. The length of line joining the centers of the surfaces is $\sqrt{(8/12)^2 + (4/12)^2} = 0.745$ ft. The surfaces being parallel, $\cos \phi_1 = \cos \phi_2 = (8/12)/0.745 = 0.898$. The area of the pan $A_2 = (9/12)^2$ or 0.5625 sq ft.

$$F_1 = \frac{0.898 \times 0.898 \times 0.5625}{3.1416 \times 0.745^2} = 0.26$$

The emissivity of the steel top, from the data of Table 9.3 for oxidized steel, is estimated at 0.8. The absorptivity of the cake is estimated to be 0.9. $T_1 = 500 + 460 = 960$. $T_2 = 160 + 460 = 620$. From equation 9.57

$$q = 0.8 \times 0.9 \times (18 \times 20)/144 \times 0.26 \times 0.173 \times 10^{-8}(960^4 - 620^4)$$
$$= 568 \text{ Btu per hr}$$

Note that this is a heat rate, at the cake, of 568/0.5625 or 1010 Btu per (hr sq ft). The equivalent radiation conductance is thus 1010/(500 − 160) or 2.97 Btu per (hr sq ft °F).

The accuracy of the approximation for the shape factor could be improved by dividing the area into a number of subareas and finding the shape factor of each.

Shape factors for a number of simple plane figures such as parallel plane discs, squares and rectangles, adjacent rectangles in perpendicular planes, and an element dA and a parallel rectangle have been published by Hottel [12] and by Moon.[13] A simple mechanical integrator for irregular objects, devised by Hottel, was used by Raber and Hutchinson [16] to measure shape factors for standing and seated persons.

Where surface areas are large with respect to the distance between them, i.e., r^2/A approaches zero, multiple reflections must be considered. Instead of the product of the emissivities that appears in equation 9.57, the emissivity factor in equation 9.49 should be used.

Calculation is often simplified by treating radiation with an equivalent surface conductance, defined by

$$q = h_r A(T_1 - T_2) = A\epsilon\sigma(T_1{}^4 - T_2{}^4) \tag{9.62}$$

from which

$$h_r = 4\epsilon\sigma\left(\frac{T_1 + T_2}{2}\right)^3 \left\{1 + \left[\frac{T_1 - T_2}{T_1 + T_2}\right]^2\right\} \tag{9.62a}$$

$$\cong 0.00692\epsilon(T_m/100)^3 \tag{9.62b}$$

Example. Find the equivalent surface conductance for radiation from a steel pipe at 330°F to surroundings at 70°F, if the emissivity of the rough steel surface is 0.95.

$$T_1 = 330 + 460 = 790°\text{R}$$

$$T_2 = 70 + 460 = 530°\text{R}$$

$$T_m = (790 + 530)/2 = 660°\text{R}$$

From equation 9.62*a*

$$h_r = 4 \times 0.95 \times 0.173 \times 10^{-8} \times 660^3 \left\{1 + \left[\frac{790 - 530}{790 + 530}\right]^2\right\}$$

$$= 0.00692 \times 0.95 \times 6.6^3(1 + 0.0388)$$

$$= 1.96 \text{ Btu per (hr sq ft °F)}$$

9.21. Heat Balance, Radiation Included. A steady-state radiation phenomenon requires that the heat involved be transferred from a source or to a sink as sensible heat, as latent heat, or as electrical or chemical heat. For example, a sample of grain being dried by an infrared lamp is receiving radiant energy at a much faster rate than it is emitting it. The net receipt is dissipated as latent heat for evaporating the moisture to be removed, plus heat conducted from the surface through the mass and heat from the surface as convected heat. Mathematically this is

$$q = w(h_g - h_f) - kA(dt/dx) + h_cA(t_1 - t_3) \tag{9.63}$$

where q = net radiant income, Btu per hr.
w = drying rate, lb moisture per hr.
h_g = heat content of the water vapor at the temperature at which it leaves the system.
h_f = heat content of the moisture to be evaporated at t_1.
h_c = convective-heat-transfer coefficient, surface at t_1 to passing air at t_3.

Note that the dependent variables are h_g and h_f which are related to t_1 and t_3. The surface temperature t_1 in equation 9.63 (T_2, equation 9.57) is dependent upon the irradiation and upon the dissipating ability of the system as defined by equation 9.49. Note that equation 9.63 can operate as written or by any combination of parts. Radiation received by a wall is dissipated by conduction and convection. An object suspended by a thin wire would dissipate net radiation by convection only.

HEAT EXCHANGERS

A heat exchanger is a device for transferring heat from a hot stream of fluid to a cold stream. The fluids are prevented from mixing with each other by a heat-conducting partition such as a pipe wall. Examples include refrigeration evaporators and con-

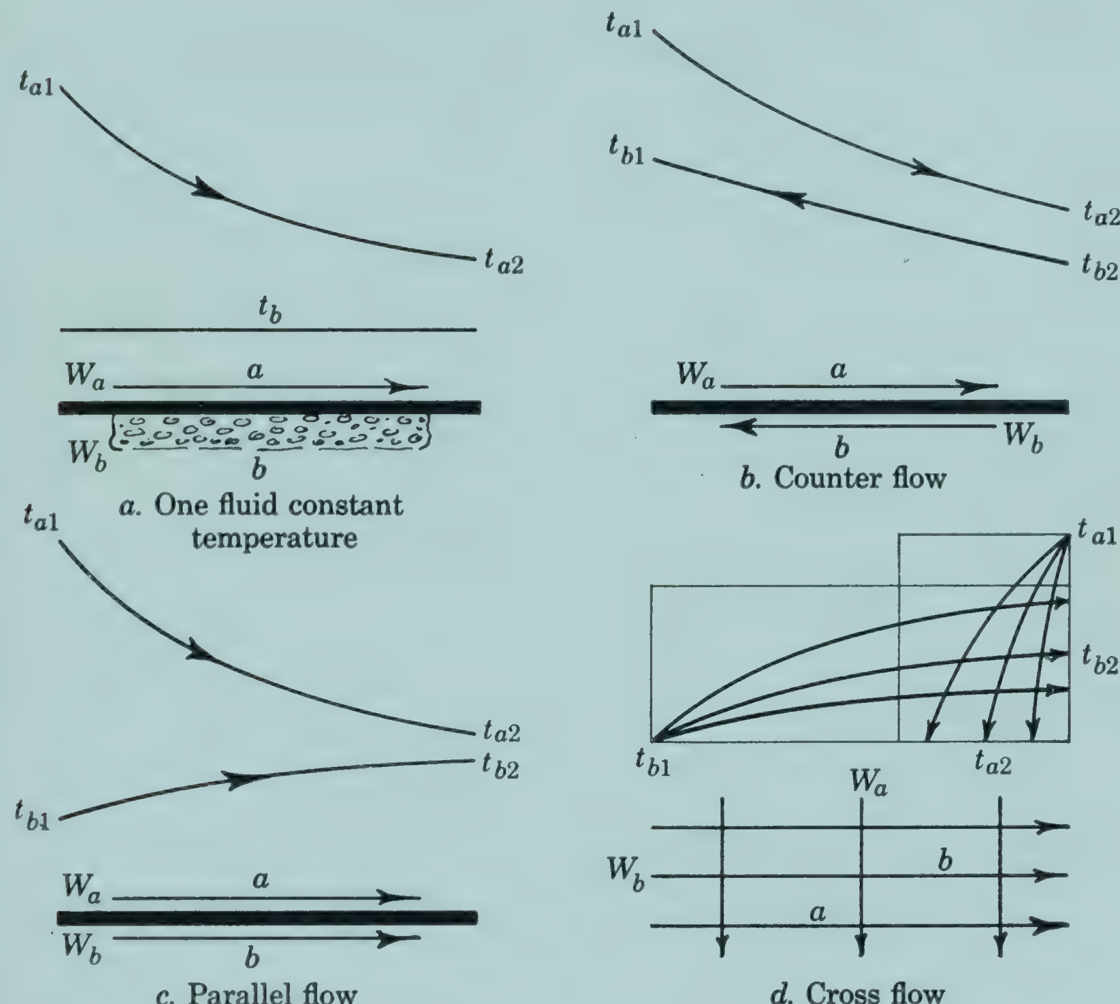


Fig. 9.8. Types of heat exchangers.

densers, automotive radiators (in reality convectors), and continuous milk pasteurizers and coolers. Of the many industrial arrangements, those that warrant discussion here are diagrammed in Fig. 9.8.

9.22. Heat-Exchanger Types. In Fig. 9.8, subscript a denotes the fluid to be heated or cooled, whereas b is for the heating or cooling medium. Subscript 1 denotes the position where fluid a enters, and subscript 2, the position where it leaves. The fluid rate, in pounds per hour is given by w, and the heat capacity by c, so that cw is the heat-capacity rate in Btu per (hr °F). The

over-all unit conductance from fluid *a* to fluid *b* is *U* Btu per (hr sq ft °F).

The exchanger where one fluid is constant in temperature is shown in Fig. 9.8*a*. Fluid *b* is constant in temperature because it is undergoing a change of state, gaining or losing energy as it evaporates or condenses. This is also a special case where fluid *b*, without changing state, has such a high rate of flow that it experiences practically no change in temperature.

In the counterflow exchanger, Fig. 9.8*b*, fluids *a* and *b* move in opposite directions. Fluid *a* can thus be brought nearly to the temperature at which fluid *b* enters if enough surface area is provided.

In the parallel-flow exchanger, Fig. 9.8*c*, with both fluids moving in the same direction, fluid *a* cannot possibly be brought to the entering temperature of fluid *b*. The mean temperature difference is obviously smaller in the parallel-flow arrangement than in counterflow, so that a greater surface area is required. This arrangement is therefore seldom used.

In the cross-flow exchanger, two fluids move in a number of separate parallel channels arranged so that the streams of fluid *a* cross those of fluid *b*, as in the automobile radiator. The several streams of fluid *a* do not mix with each other until after leaving the heat-exchange surface, as is also true for fluid *b*. This arrangement is less effective than counterflow, but better than parallel flow. It is used because of convenience in providing for the supply and removal of the fluid streams at relatively large rates of flow and short traverses of surface.

9.23. Heat-Exchanger Analysis, One Fluid Constant in Temperature. Analysis of temperature differences in a heat exchanger can be illustrated by the type in Fig. 9.8*a* with one fluid constant in temperature. Idealizations required here are:

1. The fluids gain or lose no heat, except through the transfer surface.
2. The specific heat and over-all thermal conductance do not change with temperature.
3. The fluid is completely mixed at any point, so that its temperature at any point is uniform.
4. Heat is not conducted in the direction of fluid flow by the walls nor by the fluids.

5. No leakage of fluid occurs, therefore the rate of flow is the same at any element of the surface.

With these idealizations, the heat which is transferred through an element of surface dA causes a change in temperature of fluid a,

$$-c_a w_a \, dt_a = U(t_a - t_b) \, dA \tag{9.64}$$

The negative sign indicates that fluid a drops in temperature when t_a is higher than t_b. Separating the variables for integration between definite limits

$$\int_{t_{a1}}^{t_{a2}} \frac{dt_a}{t_a - t_b} = \int_0^{A_2} \frac{-U \, dA}{c_a w_a} \tag{9.65}$$

With t_b, U, and $c_a w_a$ constant, this readily integrates to

$$\ln (t_{a2} - t_b)/(t_{a1} - t_b) = UA_2/c_a w_a \tag{9.66}$$

The dimensionless ratio in equation 9.66 of UA_2, the over-all conductance or heat-transferring ability per degree temperature difference, to $c_a w_a$, the heat-capacity rate or heat rate required per hour per degree Fahrenheit change in fluid temperature is a significant criterion for transfer in heat exchangers.

From equation 9.66

$$\frac{t_{a2} - t_b}{t_{a1} - t_b} = e^{\frac{-UA_2}{c_a w_a}} \tag{9.67}$$

Equation 9.67 can be recognized as the expression for the "by-pass" factor for an air-cooling coil discussed in Section 10.15.

Example. Find the temperature to which milk will be cooled by a direct-expansion surface cooler with a constant refrigerant temperature of 33°F if the milk rate is 1 gpm (516 lb per hr), the initial temperature is 80°F, the cooler area is 10 sq ft, and the over-all thermal conductance is 110 Btu per (hr sq ft °F). The specific heat of milk is 0.93 Btu per (lb °F).

$$\frac{t_{a2} - 33}{80 - 33} = \frac{1}{e^{\frac{110 \times 10}{0.93 \times 516}}} = \frac{1}{e^{2.28}} = \frac{1}{9.75}$$

$$t_{a2} = 33 + (80 - 33)/9.75 = 33 + 4.82 = 37.82°\text{F}$$

The effectiveness, or change in temperature in proportion to the maximum possible change, is a useful characteristic of a heat-

exchange system. Denoting the effectiveness for this case, where one fluid is constant in temperature, as E_0.

$$E_0 = (t_{a1} - t_{a2})/(t_{a1} - t_b) \tag{9.68}$$

Note that

$$t_{a1} - t_{a2} = (t_{a1} - t_b) - (t_{a2} - t_b) \tag{9.69}$$

and substituting from equation 9.67 into 9.69, and then into 9.68,

$$E_0 = 1 - e^{\frac{-UA_2}{c_a w_a}} \tag{9.70}$$

The mean temperature difference, Δt_m often useful in heat-exchange calculations, is defined by

$$\Delta t_m \, UA_2 = c_a w_a (t_{a1} - t_{a2}) \tag{9.71}$$

Solving for Δt_m in 9.71 and then substituting the value of $UA_2/c_a w_a$ from equation 9.66

$$\Delta t_m = \frac{(t_{a1} - t_b) - (t_{a2} - t_b)}{\ln (t_{a1} - t_b/t_{a2} - t_b)} \tag{9.72}$$

This mean temperature difference is seen to be a logarithmic mean of the initial and final differences.

9.24. Analysis of Counter- and Parallel-Flow Exchangers. Analytic expressions for the counterflow and parallel-flow exchangers can be derived in a manner similar to the procedure used in the previous section. In these cases t_b is not constant, but an expression for it can be found from t_a, the fluid heat-capacity rates and appropriate terminal temperatures. Integration of the differential equations yields, for the counterflow effectiveness E_c (see Fig. 9.9 for notation).

$$E_c = \frac{(t_{a1} - t_{a2})}{(t_{a1} - t_{b2})} = \frac{1 - e^{-[(1 - 1/R)UA_2/c_a w_a]}}{1 - \dfrac{1}{R} e^{-[(1 - 1/R)UA_2/c_a w_a]}} \tag{9.73}$$

In equation 9.73, R is the ratio of the heat-capacity rates, $c_b w_b/c_a w_a$. R is usually equal to or greater than 1. When $R = 1$, equation 9.73 becomes indeterminate. For this special case, the solution is

$$E_c = \frac{UA_2/c_a w_a}{UA_2/c_a w_a + 1} \tag{9.74}$$

Curves for values of E_c are given in Fig. 9.9.

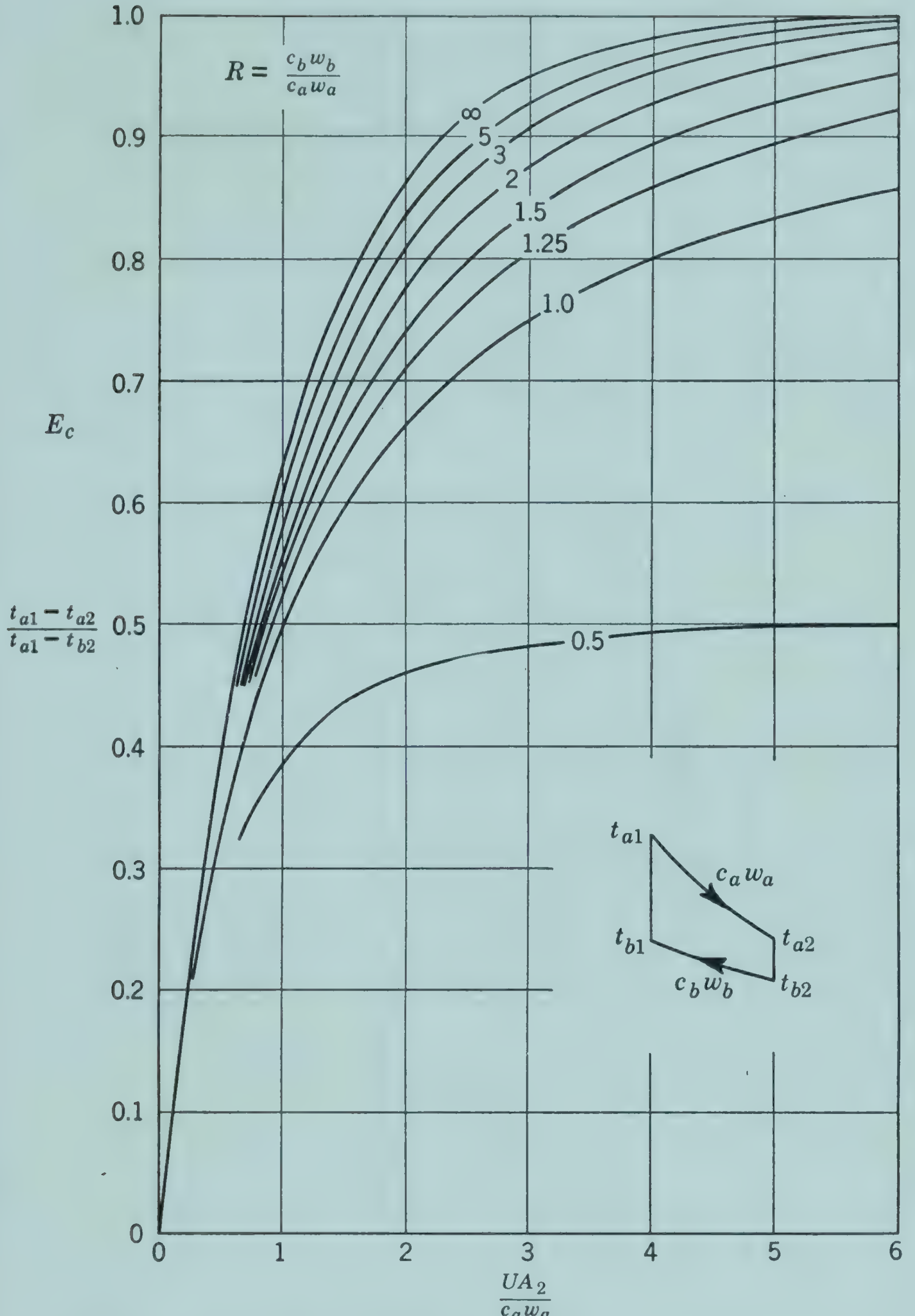


Fig. 9.9. Effectiveness curves for counterflow heat exchangers.

For the parallel-flow exchanger, effectiveness E_p,

$$E_p = \frac{(t_{a1} - t_{a2})}{(t_{a1} - t_{b1})} = \frac{1 - e^{-[(1 + 1/R) UA_2/c_a w_a]}}{1 + 1/R} \tag{9.75}$$

Curves for values of E_p are given in Fig. 9.10.

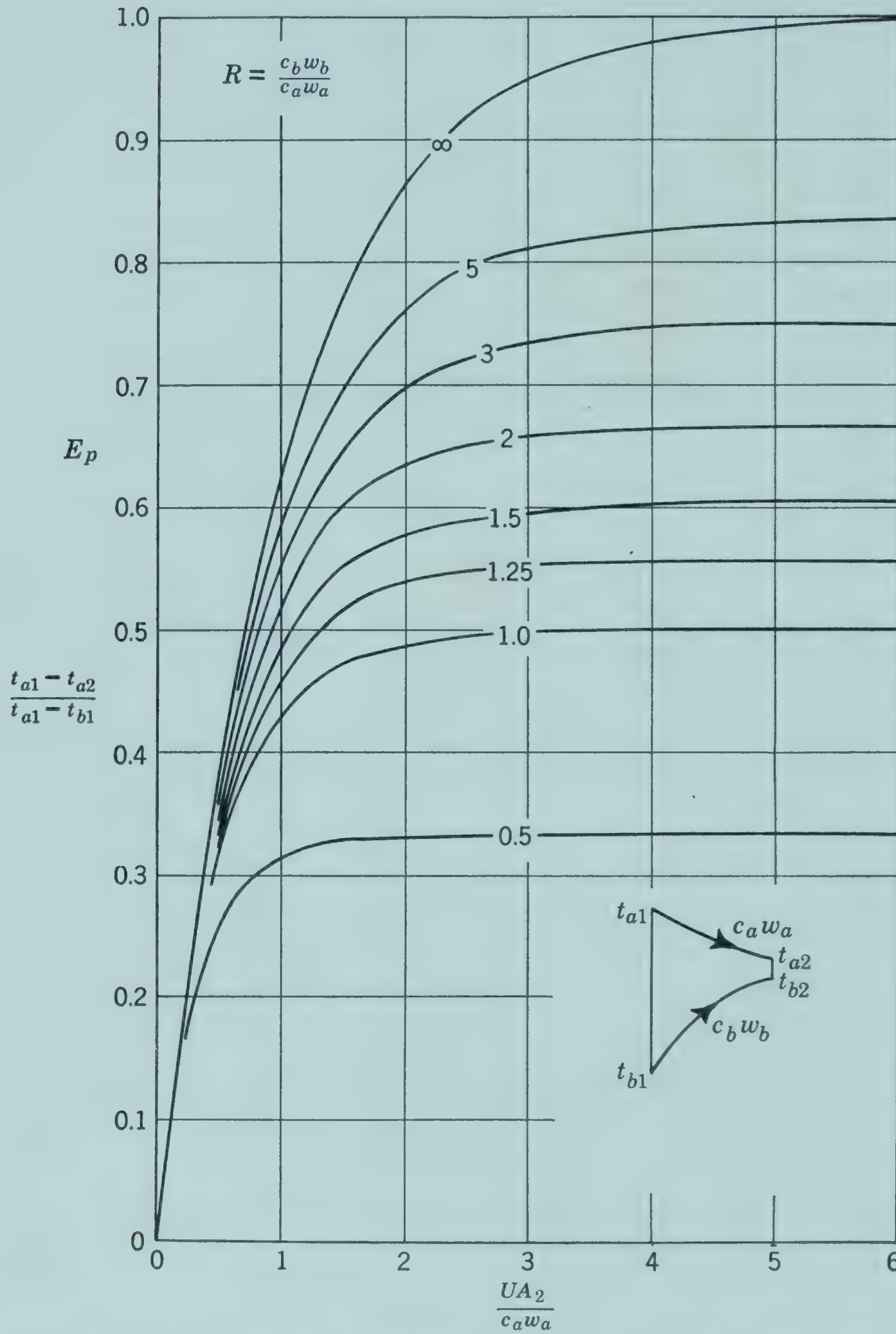


Fig. 9.10. Effectiveness curves for parallel-flow heat exchangers.

For both counterflow and parallel-flow exchangers, the mean temperature difference can be shown to be a logarithmic mean difference of the terminal temperatures. The logarithmic mean temperature difference is useful in design problems where the area required to meet specified temperatures can be found by equation

9.71. However when the area is known and the temperature that will result from given initial temperatures and flow rates is to be calculated, a trial-and-error procedure is required with equation 9.71, since, after inserting the logarithmic mean difference, it is not explicit in t_{a2}. The effectiveness curves can be used for a direct solution of t_{a2}.

Example. Find the cooler surface area required for cooling 1 gpm (516 lb per hr) of milk from 80 to 36°F, over a surface cooler, using chilled water entering at 33°F, at three times the milk rate, if the over-all thermal conductance is 110 Btu per (hr sq ft °F). The specific heat of milk is 0.93 Btu per (lb °F).

$$R = (1 \times [3] \times 516)/(0.93 \times 516) = 3.226$$

$$t_{b1} = 33 + (80 - 36)/3.226 = 33 + 13.63 = 46.63$$

$$\Delta t_m = \frac{(80 - 46.63) - (36 - 33)}{\ln (80 - 46.63)/(36 - 33)} = \frac{33.37 - 3}{\ln 33.37/3} = \frac{30.37}{\ln 11.12} = 12.6$$

$$q = (0.93 \times 516)(80 - 36) = 21{,}100 \text{ Btu per hr}$$

$$A = \frac{q}{U\,\Delta t_m} = 21{,}100/(110 \times 12.6) = 15.2 \text{ sq ft}$$

Example. Find the temperature to which milk will be cooled at a milk rate of 1 gpm (516 lb/hr), the initial temperature 80°F, by a counterflow surface cooler with 10 sq ft of surface, supplied with chilled water at 33°F, at three times the milk rate. A thermal conductance U of 110 Btu per (hr sq ft °F) is expected.

$$R = (1 \times 3 \times 516)/(0.93 \times 516) = 3.226$$

$$UA_2/c_a w_a = (110 \times 10)/(0.93 \times 516) = 2.285$$

From Fig. 9.9 at $UA_2/c_a w_a = 2.285$ and interpolating for $R = 3.226$, find

$$E_c = (t_{a1} - t_{a2})/(t_{a1} - t_{b2}) = 0.85$$

$$t_{a2} = t_{a1} - 0.85(80 - 33)$$

$$= 80 - 40 = 40$$

9.25. Analysis of Cross-Flow Exchangers. For the cross-flow exchanger, the analytic solution is more involved, since temperatures of both fluid a and fluid b vary with both x and y as illustrated in Fig. 9.11. Fortunately the effectiveness of the cross-flow exchanger E_x can be expressed by the same dimensionless

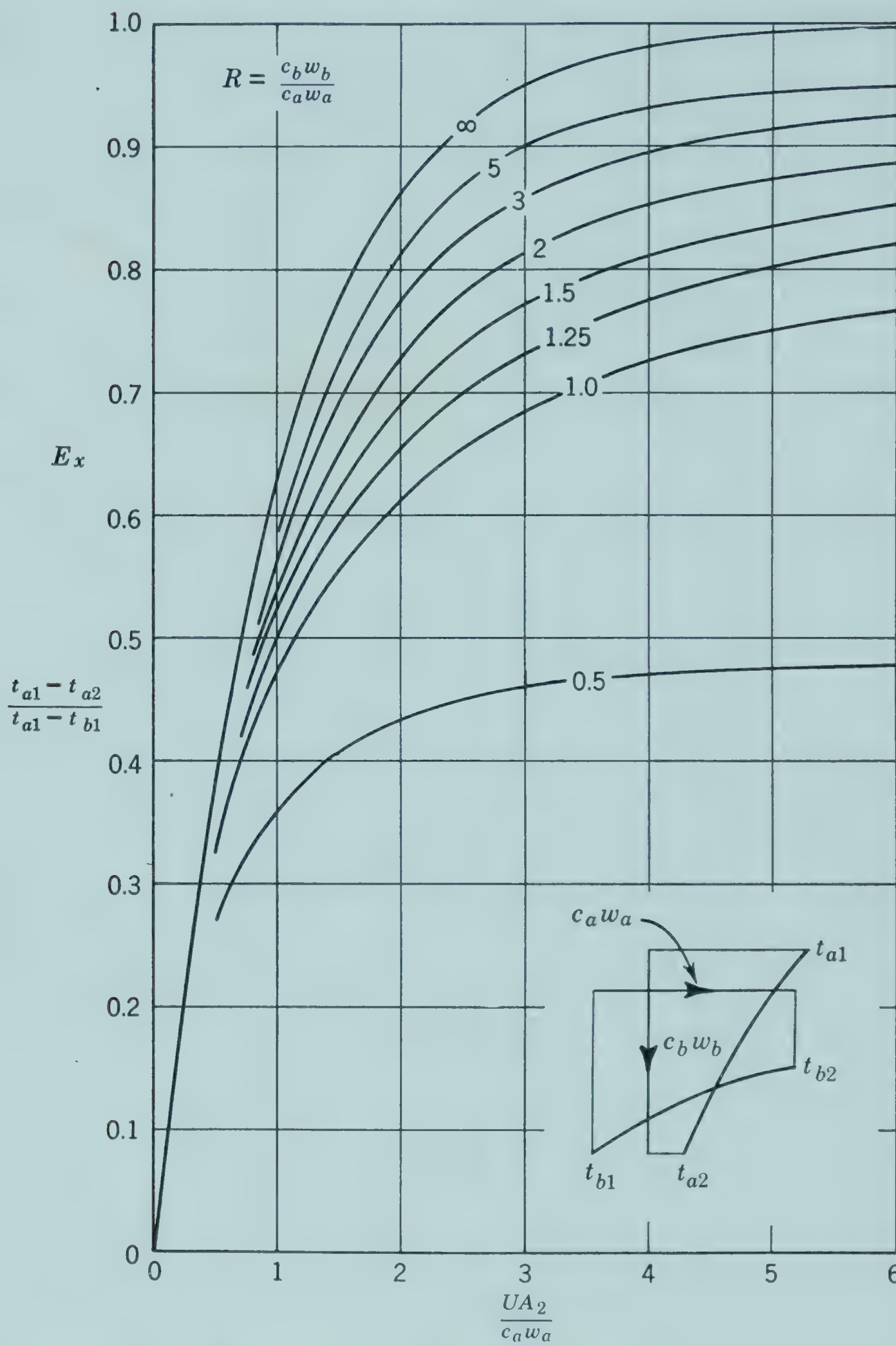


Fig. 9.11. Effectiveness curves for cross-flow heat exchangers.

parameters $UA_2/c_a w_a$ and R as the simpler heat exchangers. Values [*,15] of E_x given in Fig. 9.11 indicate that the cross-flow exchanger is between the counterflow and parallel-flow exchangers in effectiveness.

* Personal communication. R. Seban, University of California, 1952.

REFERENCES

1. Boelter, L. M. K., et al. *Heat Transfer Notes.* University of California Press. 1946.
2. Brown, A. I., and Marco, S. M. *Introduction to Heat Transfer.* McGraw-Hill. 1942.
3. Carslaw, H. S., and Jaeger, J. C. *Conduction of Heat in Solids.* Oxford. 1947.
4. Dunkle, R. V., et al. *Thermal Radiation Project 10 Report,* Code NR015–202. Univ. of Calif. Inst. Eng. Res. 1950.
5. Dusinberre, G. M. *Numerical Analysis of Heat Flow.* McGraw-Hill. 1949.
6. Emmons, H. W. Numerical Solution of Partial Differential Equations. *Quarterly of Applied Mathematics* 2:173–195. 1944.
7. Gier, J. T., et al. Absolute Spectral Reflectivity of Certain Pigments and Metals in the Wavelength Range Between 2 and 15 Microns. *Thermal Radiation Project 7 Report,* Code NR015–202. Univ. of Calif. Inst. of Eng. Res. 1949.
8. *Heating, Ventilating, and Air Conditioning Guide.* Am. Soc. of Heating and Ventilating Engrs. 1950.
9. *International Critical Tables* 5:253. McGraw-Hill. 1929.
10. Jakob, Max. *Heat Transfer.* Vol. 1. Wiley. 1949.
11. Kern, D. Q. *Process Heat Transfer.* McGraw-Hill. 1950.
12. McAdams, W. H. *Heat Transmission.* Second Edition. McGraw-Hill. 1942.
13. Moon, Parry. *Scientific Basis of Illuminating Engineering.* McGraw-Hill. 1936.
14. Newman, A. B. Drying of Porous Solids: Diffusion and Surface Emission Equations. *Trans. Am. Inst. Chem. Engrs.* 27:212–214. 1931.
15. Nusselt, W. A New Heat Transfer Formula for Cross Flow. *Technischke Mechanik Thermodynamik 1,* No. 12. 1930.
16. Raber, B. F., and Hutchinson, F. W. Optimum Surface Distribution in Panel Heating and Cooling Systems. *Heating, Piping, Air Conditioning* 15:602–614. 1943.
17. Southwell, R. V. *Relaxation Methods in Engineering Science.* Oxford. 1940.

PROBLEMS

1. Find the rate of heat loss per sq ft from the hot-air supply duct of a spray drying chamber. The duct is insulated with a 2-in. layer of glass-fiber batts. The inner surface temperature is 300°F, the outer, 75°F. Thermal conductivity k of the batts = 0.025 Btu per (hr sq ft °F per ft).
2. A heat exchanger built with 16-gage copper tubes has an over-all unit conductance of 300 Btu per (hr sq ft °F) under certain operating con-

ditions. Predict the over-all conductance if 18-8 stainless steel is used instead of copper in a new design.

3. A furnace wall is to be built of firebrick 8 in. thick and building brick of the same thickness. The thermal conductivities are 0.9 and 0.4 Btu per (hr sq ft °F per ft) respectively. The inner surface of the firebrick is at 1800°F, the outer surface of the structural brick at 90°F. Find the heat rate per square foot and the temperature of the brick interface. What fraction of the resistance is provided by the firebrick?
4. A number 2 can containing 1.25 lb of a liquid which has a specific heat of 0.9 Btu per lb °F and a density of 64 lb per cu ft will be rotated during processing in such a way that the thermal conductance from the contents to the can will be 10 Btu per (hr sq ft °F). The temperature within the can will be uniform at any instant, except for the laminar layer at the surface. The outside surface conductance, in the steam atmosphere of the retort, will be 1000 Btu per (hr sq ft °F). The can, with its contents initially at 160°F, is placed in a retort where the temperature is promptly raised to 240°F. Find the temperature in the can at 5, 10, and 15 min.
5. A rectangular can, $2\frac{7}{8} \times 3\frac{3}{8} \times 6$ in., contains a food product similar to that in the example of sect. 9.7. Find the time for the center to reach 235°F when the can is at the same initial temperature and is then subject to the same retort temperature as in the example for the cylindrical can in problem 4.
6. Estimate the time required to bring the center of a spherical melon, 9 in. in diameter, to a temperature of 42°F after the melon, initially at 80°F, is placed into a refrigerator at 35°F. The surface thermal conductance will be 5 Btu per (hr sq ft °F), the density 62.4 lb per cu ft, the specific heat 0.90, and the thermal conductivity 0.25 Btu per (hr sq ft °F per ft). Also estimate the mean temperature of the melon when the center has reached 42°F.
7. Estimate the heat loss rate by convection from the top of a blancher box, 6 ft wide and 36 ft long, in a room at 70°F. The cover plates are aluminum-painted galvanized iron, uninsulated. The box is direct steam-heated at 210°F.
8. What wattage per lineal foot can be used for a direct-immersion electric heater, $\frac{3}{8}$ in. outside diameter, to be placed horizontally in a tank for heating molasses? It is specified that the surface temperature of the heater must not rise over 180°F. The specific heat at a water content of 22.5 per cent is 0.46 Btu per (lb °F). The thermal conductivity is estimated from data on sucrose to be 0.19 Btu per (hr sq ft °F per ft) at 68°F.
9. Estimate the surface thermal conductance for heating tomato pulp from 60 to 170°F at a velocity of 4 ft per sec in 1-in.-diameter tubes (0.88 in. ID). The tube-wall temperature, in counterflow, will be 120°F at one end and 180°F at the other. Data on tomato pulp are given in the example in sect. 9.13.
10. Estimate the rate of heat loss by radiation from the blancher box cover of problem 8, if the mean wall and ceiling temperature is 75°F.

11. Find the heat loss rate by radiation from a furnace tube, 3 ft in diameter and 10 ft long, which is in a concrete-block dehydrater furnace chamber, 6 ft wide and 7 ft high. The tube-surface temperature is 450°F, and the chamber-wall temperature averages 180°F.
12. Find the thickness of cork insulation which must be used on a 1½-in. horizontal pipe, carrying brine at −10°F, in order to avoid condensation on the surface of the insulation when the room temperature is 70°F and the dew point is 60°F. The brine inside-surface-conductance is 200 Btu per (hr sq ft per °F). The air moves by free convection.
13. Find the length of 0.88-in. ID tube required for heating 3900 lb per hr of tomato pulp from 60 to 170°F, in a counterflow exchanger with hot water entering at 186°F and leaving at 156°F. The over-all unit conductance U is estimated at 250 Btu per (hr sq ft °F).
14. A finned-tube refrigeration coil is rated at 2400 Btu per hr at 10°F difference between *initial* air temperature and refrigerant, at an air rate of 400 cu ft per min. Find the exit air-temperature and the by-pass factor. Also find the over-all conductance UA_2 and the heat rate per degree *mean* temperature difference.
15. To increase capacity, the air rate through the coil in problem 14 is to be raised to 600 cu ft per min. Predict the new value of UA_2, if 70 per cent of the over-all air to refrigerant thermal resistance was from air to surface in problem 14. Find the new exit-air to refrigerant temperature difference and the coil capacity, in Btu per hr per degree initial difference. Also find the new by-pass factor.

CHAPTER 10

Air-Vapor Mixtures (The Psychrometric Chart)

NOMENCLATURE

A = area, sq ft.
B = by-pass factor, dimensionless.
f_v = surface vapor conductance, lb per (hr sq ft lb per sq ft).
f = surface heat transfer, Btu per (hr sq ft °F).
H = absolute humidity, lb moisture per lb dry air.
h_a = heat content of an atmospheric mixture, Btu per lb dry air.
h_{fg} = latent heat of evaporation, Btu per lb.
h_g = heat content of water vapor, Btu per lb.
M = molecular weight.
N = number of pound moles, weight in pounds divided by molecular weight.
p = pressure, lb per sq in. absolute.
p_a = pressure exerted by dry air, lb per sq in.
p_{at} = atmospheric pressure, lb per sq in. = $p_a + p_v$.
p_s = pressure exerted by the water vapor in the atmosphere when saturated, lb per sq in.
p_v = pressure exerted by the water vapor in the atmosphere when unsaturated, lb per sq in.
q = heat rate, Btu per hr.
R = universal gas constant = 1545, ft lb per (lb mole °R).
T = absolute temperature, °R = °F + 460.
t = temperature, °F.
t_a = air temperature, °F.
t_w = wet-bulb temperature, °F.
V = gas volume, cu ft.
v = humid volume, cu ft per lb dry air.
v_{as} = volume of 1 lb dry air at saturated conditions, cu ft.
v_{vs} = volume of 1 lb water vapor at saturated conditions, cu ft.
W = quantity of gas, lb.
w = evaporation rate, lb water per hr.

In the field of processing, air is used as a heat-transfer medium, a source or sink for water vapor, a source of oxygen for combus-

tion, and a vehicle for vapors which are to be removed as undesirables or used as processing media.

Dry air at sea level has a percentage volumetric composition of: N_2, 78.03; O_2, 20.99; A, 0.94; CO_2, 0.03; H_2, 0.01; Ne, 0.00123; He, 0.0004; Kr, 0,00005; Xe, 0.000006. For engineering purposes, air is considered composed of nitrogen and oxygen; the tabulated data apply at sea level, atmospheric pressure, 14.7 lb per sq in.

	Composition		
	By Volume	*By Weight*	*Molecular Weight*
Nitrogen	79%	76.8%	28.02
Oxygen	21%	23.2%	32.00
Air (dry)	——	——	28.97

THE LAWS

The pressure, volume, weight, and thermal properties of a single gas or a mixture of gases are related by a number of mathematical formulas or laws. These laws hold satisfactorily for moderate or normal processing conditions. For high pressures, a number of atmospheres, and high temperatures, the deviation from these laws must be considered if accurate results are expected.

10.1. The Ideal Gas Law. This law has two forms,

$$pV = NRT \tag{10.1}$$

and

$$p_1V_1/T_1 = p_2V_2/T_2 \tag{10.2}$$

where p = pressure, lb per sq ft absolute.
V = gas volume, cu ft.
T = absolute temperature, °R (°F + 460).
N = number of pound-moles = weight of gas, lb, divided by its molecular weight = W/M.
R = universal gas constant = 1545 ft lb per (lb mole °R).

Equation 10.2 may be used for any consistent set of terms since it is based upon a ratio. Equation 10.1 must be used with the term dimensions shown since the gas constant is based upon these dimensions. The ideal gas law does not hold perfectly for extreme conditions. It is entirely suitable for the usual processing calculations, however.

10.2. Amagat's Law. The volume of a mixture of gases at a certain temperature and pressure is equal to the sum of the volumes of the individual gases at the same conditions, or:

$$V = V_1 + V_2 + V_3 + \cdots \tag{10.3}$$

10.3. Dalton's Law. Each component in a mixture of gases exerts the same pressure it would exert if it alone occupied the same volume at the same temperature, or:

$$p = p_1 + p_2 + p_3 + \cdots \tag{10.4}$$

The weight of the mixture is, of course, the sum of the weights of the components, e.g.:

$$W = W_1 + W_2 + W_3 + \cdots \tag{10.5}$$

Using the gas law, the general expression for V is,

$$V = WRT/pM \tag{10.6}$$

and for a mixture confined in a space of V

$$\frac{WRT}{pM} = \frac{W_1RT}{p_1M_1} = \frac{W_2RT}{p_2M_2} = \frac{W_3RT}{p_3M_3} = \cdots \tag{10.7}$$

or

$$\frac{W}{pM} = \frac{W_1}{p_1M_1} = \frac{W_2}{p_2M_2} = \frac{W_3}{p_3M_3} = \cdots \tag{10.8}$$

A pound-mole of gas M occupies 359 cu ft at atmospheric pressure and 32°F.

The usefulness of these formulas can be demonstrated by the following examples.

Example 1. A 75-cu-ft pressure tank contains dry air at 40 lb per sq in. gage and 70°F. What is the weight of the air in the tank? Using formula 10.6

$$W = \frac{(40 + 14.7)144 \times 75 \times 28.97}{1545 \times (460 + 70)} = 20.9 \text{ lb}$$

Example 2. How much nitrogen must be added to the tank of Example 1 to bring the pressure up to 60 lb per sq in.? The increase in pressure or 20 lb per sq in. is due to the nitrogen only and independent of the air, therefore formula 10.7 may be used.

$$20.9/(40 + 14.7)144 \times 28.97 = W/(20 + 14.7)144 \times 14.01$$

$$W = 6.42 \text{ lb}$$

THE PSYCHROMETRIC CHART

Normal atmospheric air is a mixture of dry air and water vapor, atmospheric air never being completely dry. The psychrometric chart is a graphic representation of the physical and thermal properties of atmospheric air.

Problems in air-vapor mixtures which include heating, cooling, humidification, dehumidification, and mixing can be solved by the psychrometric chart or by direct calculations. The psychrometric chart will be discussed and developed step by step in order to understand its mechanics and limitations and the merits of direct calculation as compared to chart solutions.

10.4. Saturation Pressure. The water vapor in the atmosphere conforms to Dalton's law and, thus, exerts a pressure independent of the dry air. Therefore, the vapor pressures for a space saturated with water vapor can be taken directly from any standard steam table. Steam table values are preferred to those calculated from the gas law since the steam table values are more accurate.

10.5. Absolute Humidity. The pounds of moisture per pound of dry air is called the absolute humidity or, by some writers, humidity. The base (1 lb of dry air) is used since it is a constant for any change of conditions, thus facilitating calculations.

Equation 10.7 is used for the calculations.

$$W = 1 \text{ lb of dry air}$$

$$p_a = p_{at} - p_v \tag{10.9}$$

where p_a = pressure exerted by the dry air, lb per sq in.
p_{at} = pressure exerted by the atmosphere, lb per sq in.
p_v = pressure exerted by the water vapor in the atmosphere, lb per sq in.

this being the pressure exerted by the dry air. p_{at} for standard atmospheric pressure is 14.7 lb per sq in; M is 28.97; M_1 is 18.02. Therefore:

$$H = \frac{p_v}{p_{at} - p_v} \cdot \frac{18.02}{28.97} = \frac{p_v}{1.605(p_{at} - p_v)} \tag{10.10a}$$

and

$$p_v = \frac{p_{at}(1.605)H}{1.605H + 1} \tag{10.10b}$$

The linear plot with humidity and temperature as the ordinate and abscissa shown in Fig. 10.1 is the base plot for the psychrometric chart. H may be expressed in pounds or grains (1 lb = 7000 grains) per pound of dry air. The more convenient

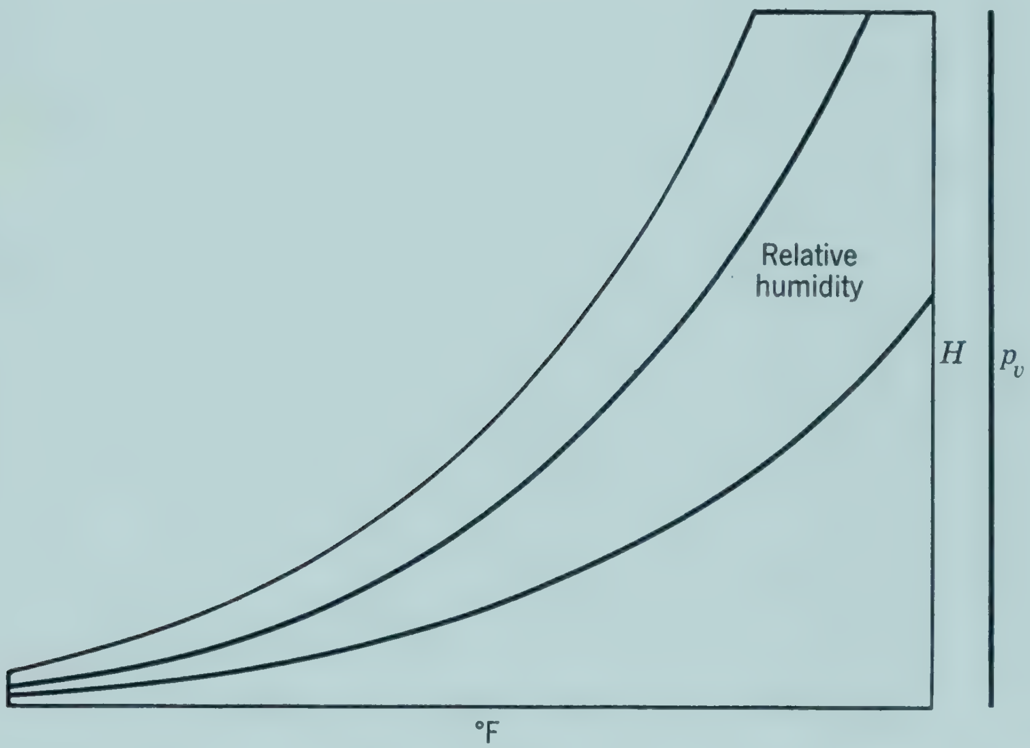


Fig. 10.1. A base plot with relative humidity and water vapor pressure.

unit will probably be used. The water-vapor pressure is calculated by equation 10.10*b* and is usually added to the plot in the position shown. Other positions are frequent, however.

10.6. Relative Humidity. Relative humidity is defined as the ratio of the actual pressure of the water vapor in the air to the pressure if the air were saturated with moisture at the same temperature. For example, if the pressure were 0.180 lb per sq ft at 70°, the relative humidity would be (0.180/0.361)100, or 50 per cent. This definition leads to a family of curves, partially shown in Fig. 10.1.

10.7. Percentage Humidity. Percentage humidity is defined as the ratio of the absolute humidity at a state to the absolute

humidity at the same temperature for a saturated condition. Relative humidity is defined as $(p/p_s)100$, percentage humidity as $(H/H_s)100$.

Although percentage humidity is more convenient for some calculations than relative humidity, relative humidity is preferable when dealing with systems where equilibrium moisture content of commodities is concerned.

10.8. Humid Volume. The humid volume of an air-water-vapor mixture is assumed (without significant error) to be the sum of the volume of 1 lb of dry air and the volume of the water vapor. The volume of 1 lb of the dry air for any temperature and pressure can be calculated from the gas law (10.1). Likewise, the volume of the water vapor *per pound of dry air* for any temperature and humidity can be calculated from the vapor pressure and humidity values previously discussed. Although vapor pressure volumes calculated thus are generally acceptable, steam table values used as shown below will yield more accurate results.

$$v = \frac{T}{T_s}\left(v_{as} + \frac{Hv_{vs}p_v}{p_{at}}\right) \tag{10.11}$$

where v = humid volume of mixture, cu ft per lb dry air at temperature T.

T = absolute temperature of the air-water-vapor mixture, R°.

T_s = absolute temperature of the air-water-vapor mixture at the saturated, dew-point temperature of the water vapor.

v_{as} = volume of 1 lb dry air at T_s and atmospheric pressure p_a; calculated from the gas law.

p_{at} = atmospheric pressure, lb per sq in.

v_{vs} = volume of 1 lb of water vapor, saturated, at the dew point, cu ft per lb (from steam table).

p_v = water-vapor pressure of the atmosphere, lb per sq in.

The isovolume lines of Fig. 10.2 were calculated by equation 10.11.

The specific volume is determined thus:

$$\text{Specific volume} = \text{Humid volume}/(1 + H) \tag{10.12}$$

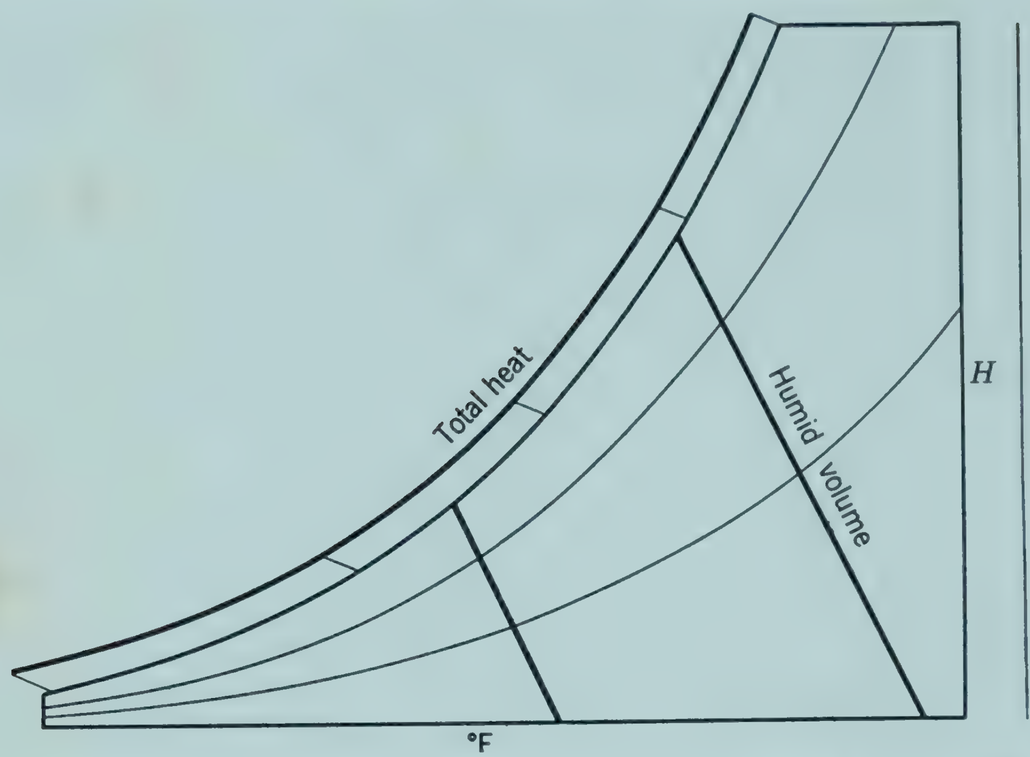


Fig. 10.2. Humid volume and total heat data are added to the base plot thus.

10.9. Total Heat, Enthalpy. The total heat or enthalpy of an air-water-vapor mixture is expressed by

$$h_a = 0.24t + Hh_g \tag{10.13}$$

where h_a = heat content of the mixture, Btu per lb of dry air, referred to zero degrees for air, and to water at 32°F for vapor.

$0.24t$ = average specific heat of dry air (0.24) times the temperature.

H = humidity.

h_g = heat content of a pound of water vapor at temperature t. This can be taken directly from a steam table or can be calculated from

$$h_g = 1075.2 + 0.45\,(t-32) \tag{10.14}$$

or

$$h_g = 1060.8 + 0.45t \tag{10.14a}$$

and

$$h_a = 0.24t + H(1060.8 + 0.45t) \tag{10.15}$$

which is sufficiently accurate for most engineering applications below a partial vapor pressure of 2 lb per sq in. The constant 1075.2 is the heat content of a pound of water vapor at 32°F;

0.45 is the specific heat of water vapor. Total heat values for saturated air are usually placed as shown in Fig. 10.2.

10.10. Adiabatic Processes. An adiabatic process is a procedure whereby there is a change from one state to another without heat exchange between system and surroundings. Consider a perfectly insulated system with a change of state from 1 to 2 as shown schematically in Fig. 10.3*a*.

The heat and mass balance is:

$$0.24t_1 + H_1(1060.8 + 0.45t_1) + (H_2 - H_1)(t_3 - 32)$$
$$= 0.24t_2 + H_2(1060.8 + 0.45t_2) \quad (10.16)$$

The water can enter the system at a temperature t_3 which can be above, below, or equal to either t_1 or t_2.

Let the subscript-2 and subscript-3 values be those at saturation. H_1 and t_1 will then define a series of lines that are placed on the psychrometric chart as shown in Fig. 10.3*b* and are called adiabatic saturation lines. They are *not* lines of constant enthalpy. The enthalpy at saturation is greater than the enthalpy at an unsaturated point on the line by the factor $(H_s - H_1)(t_s - 32)$. Calculations under 110°F are usually made on the assumption that the lines *are* constant enthalpy lines thus introducing an error equal to the heat content of the water represented by the factor noted above. The error is usually of little or no significance for conditions under 110°F but may be significant above 110°F.

Corrections may be made by calculations or by the use of adjusting lines such as those dashed on Fig. 11.12.

10.11. Wet-Bulb Temperature. Relative humidities are usually determined by observing "wet" and "dry" bulb temperatures. A laboratory thermometer with a wet gauze-covered bulb gives a wet-bulb temperature which permits relative humidity to be determined.

The evaporation of the water from the wet bulb soon attains a steady state in which heat is transferred just rapidly enough from the surroundings to provide energy for evaporation as shown in Fig. 10.3*c*.

The quantity of air passing the bulb is so great that little change in the surrounding air temperature results. This is contrasted to the usual adiabatic process where a significant air tem-

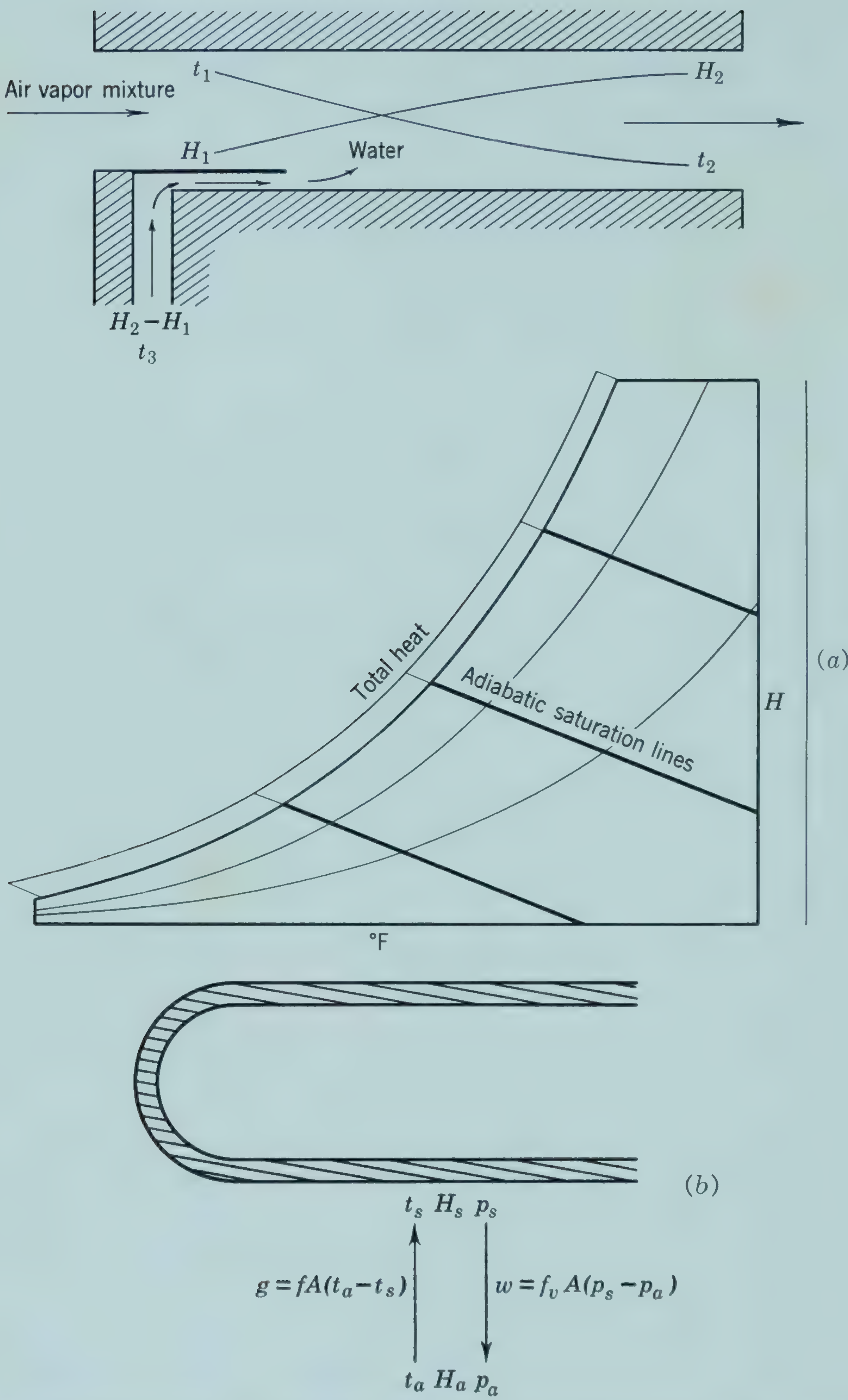


Fig. 10.3. An adiabatic process, shown schematically, is keyed to the total heat scale by adiabatic saturation lines (a). The lines represent the process when sub-2 and sub-3 values are at saturation. The wet-bulb process (b) is also represented by the adiabatic saturation lines.

perature change takes place. The wet bulb cools by evaporation of the water from the bulb, the rate being:

$$w = f_v A(p_s - p_v) \tag{10.17}$$

where w = evaporation rate, lb per hr.
f_v = surface vapor conductance, lb per (hr sq ft lb per sq ft).
A = bulb area, sq ft.
p_s = saturated pressure of water on bulb at bulb temperature (lb per sq ft).
p_v = pressure of water vapor in air.

The latent heat rate for the evaporation of equation 10.17 is:

$$q = wh_{fg} \tag{10.18}$$

where h_{fg} = latent heat of evaporation, Btu per lb.

The heat required for equation 10.18 is secured from the air as sensible heat thus:

$$q_1 = fA(t_a - t_w) \tag{10.19}$$

where f = air-film heat-transfer conductance coefficient, Btu per (hr sq ft °F).
A = wet-bulb area, sq ft.
t_a = air temperature, F°.
t_w = wet-bulb temperature, F°.

The wet bulb is of necessity cooled to such a temperature that the latent heat rate is equal to the sensible heat rate, $q = q_1$ and:

$$wh_{fg} = f_v A(p_s - p_a)h_{fg} = fA(t_a - t_w) \tag{10.20}$$

$$p_s - p_a = f(t_a - t_w)/f_v h_{fg} \tag{10.21}$$

When the vapor pressure is small as compared to the total air pressure, the pressure-humidity equation 10.10 may be simplified to

$$H_a = \frac{p_a}{1.605(14.7)} \tag{10.22}$$

and substituted in 10.21 giving

$$\frac{H_s - H_a}{t_a - t_w} = \frac{f}{1.605(14.7)f_v h_{fg}} \tag{10.23}$$

The heat and mass balance or adiabatic saturation process for the wet-bulb process is:

$$(H_s - H_a)h_{fg} = (t_a - t_s)(0.24 + 0.45H_a) \tag{10.24}$$

and

$$\frac{H_s - H_a}{t_a - t_s} = \frac{0.24 + 0.45H_a}{h_{fg}} \tag{10.25}$$

The psychrometric adiabatic-saturation line can be used as the wet-bulb line only if equations 10.22 and 10.24 are equal, that is, if:

$$\frac{f}{1.605(14.7)f_v} = 0.24 + 0.45H_a \tag{10.26}$$

Fortunately, this equation is valid for engineering problems if the following operation factors are recognized. The *rate of air* past the bulb affects the value of the coefficients f and f_v. Experience has shown that the error will be minimum for an air rate of 500 to 1000 ft per min. *Radiant heat exchange* between the wet bulb and the surroundings may be significant unless the following precautions are observed: (*a*) the bulb should be as small as practical to minimize the projected area that a radiant source or sink "sees," (*b*) the air rate past the bulb should be high so that the difference in temperature between the air and the bulb needed to compensate for radiant heat will be small. An air rate as close to 1000 ft per min as practicable is optimum, (*c*) shielding the wet bulb will eliminate the radiant heat exchange. Precaution (*c*) of itself will eliminate the radiation problem, but shielding complicates the construction of a unit. Precautions (*a*) and (*b*) combined will correct for normal radiation. All three are needed only if radiation is intense.

The most accurate values result when tables based on carefully observed data are used. Examples are Marvin's tables,[6] moist air tables by Goff and Gratch, and *Gas Tables* by Keenan and Kaye.

The validity of equation 10.26 is a result of fortuitous circumstance. It does not hold for systems composed of other materials. In other systems, air-benzene, for example, the adiabatic-saturation line and wet-bulb line do not coincide.

The adiabatic humidification line of Fig. 10.3 is frequently noted as the wet-bulb line. This nomenclature is convenient for

the casual user, but it is erroneous as regards the true properties of the line.

USES OF THE PSYCHROMETRIC CHART

10.12. State Factors. The information that can be secured from the psychrometric chart for a state condition is shown in

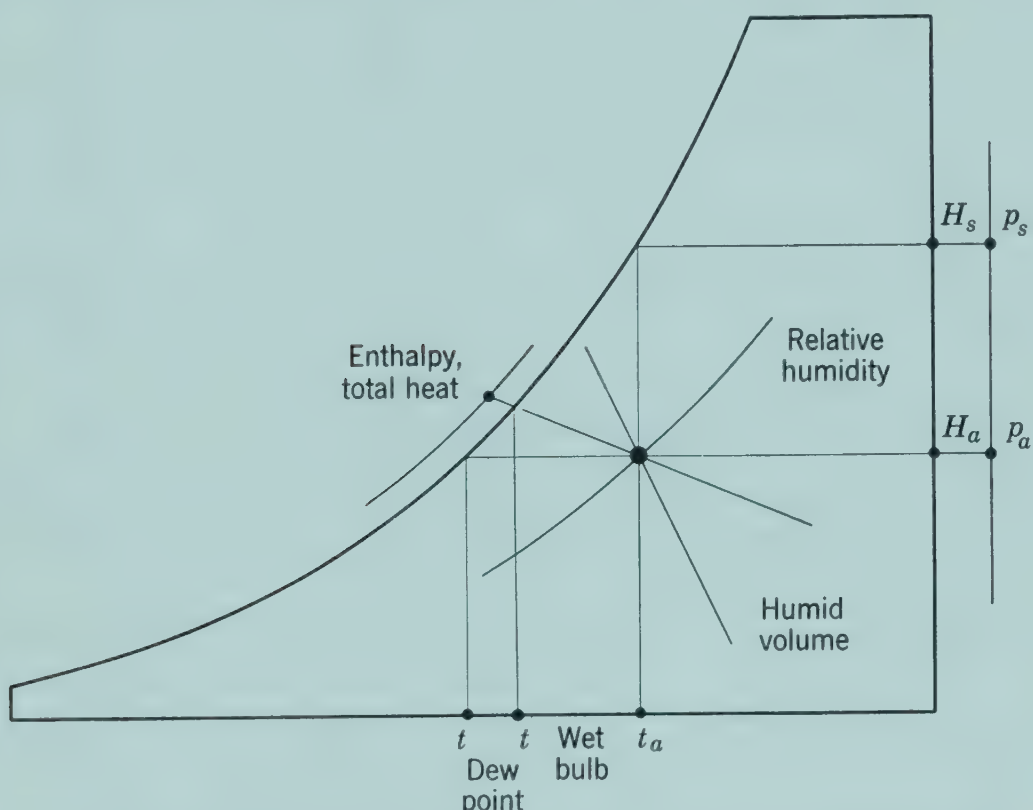


Fig. 10.4. Information that can be secured from a psychrometric chart from one state point.

Fig. 10.4. Note that the crossing of any two property lines establishes a state point from which all other values can be secured.

For example, 70°F air having a wet-bulb temperature of 55°F has a humidity H of 0.006 lb of water per lb of dry air. The dew point is 43°F; vapor pressure, 0.132 lb per sq in.; relative humidity, 37 per cent; humid volume, 13.47 cu ft dry air; and the vapor pressure, 0.361 lb per sq in. Note that the relative humidity can be calculated from the ratio p/p_s.

10.13. Cooling, Heating. Cooling or heating without changing the moisture content takes place horizontally as shown in Fig. 10.5. The heat involved per pound of dry air is $h_2 - h_1$. In cooling, the temperature of the cooling medium must be above the dew point or dehumidification will result (see section following).

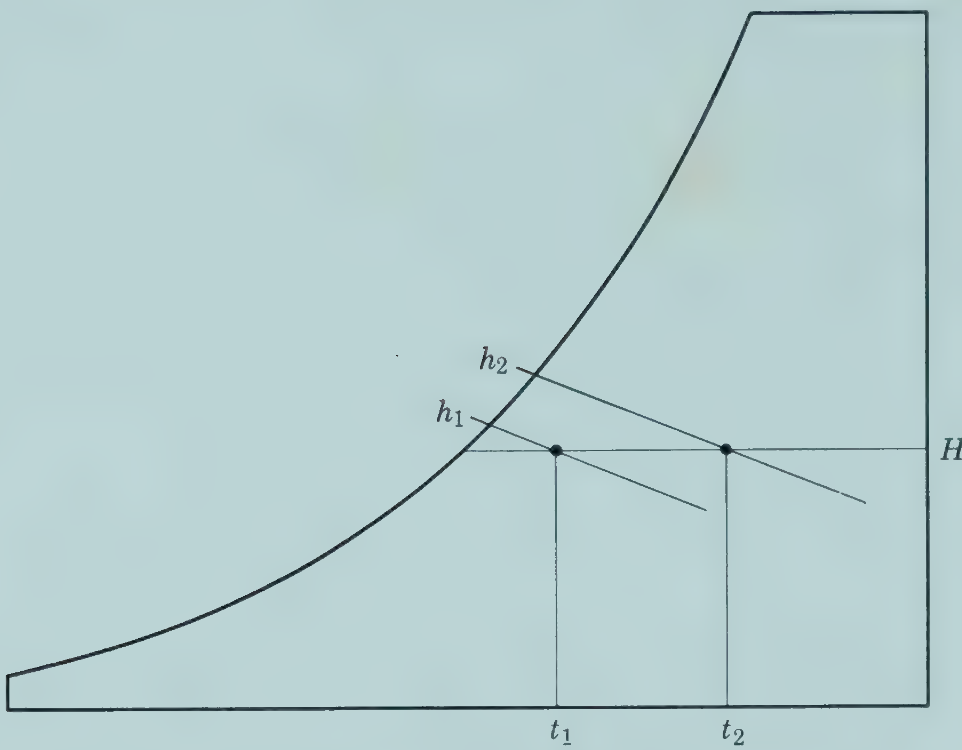


Fig. 10.5. The psychrometric heating and cooling process.

10.14. Mixtures. The state point of an air-vapor mixture resulting from mixing airs of different state points falls very

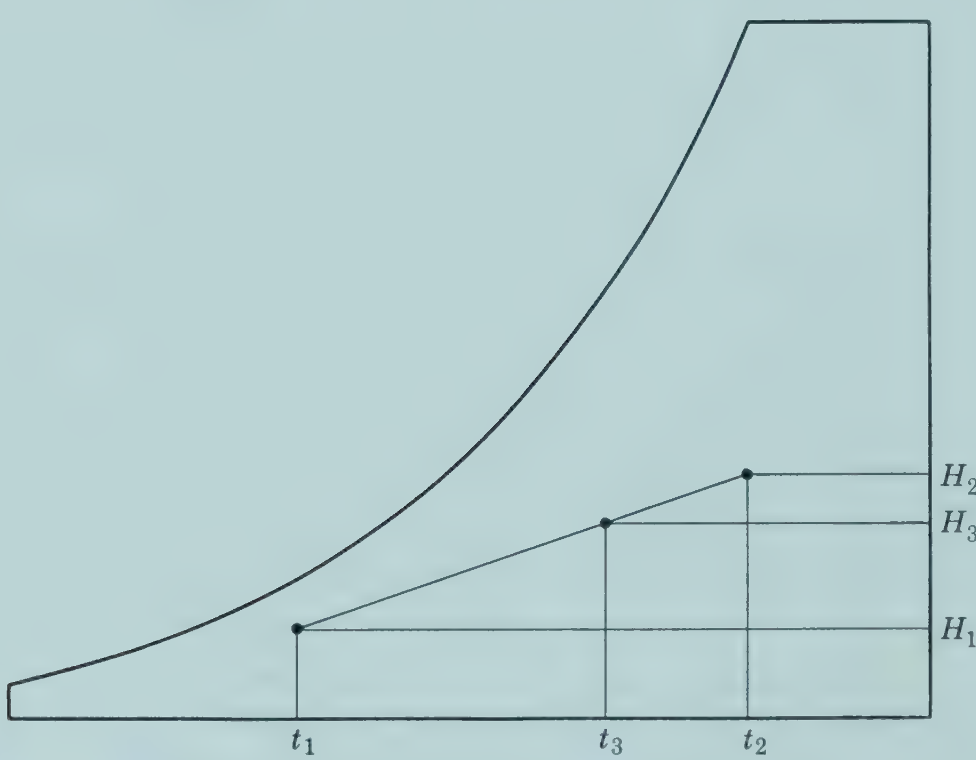


Fig. 10.6. The psychrometric process of mixing.

nearly on a straight line connecting the two initial states, Fig. 10.6. Proof of this procedure follows.

Using W as the pounds of dry air,

$$W_1 + W_2 = W_3 \tag{10.27}$$

and

$$W_1H_1 + W_2H_2 = W_3H_3 \tag{10.28}$$

$$\therefore H_3 = (W_1H_1 + W_2H_2)/(W_1 + W_2) \tag{10.29}$$

The heat required to cool W_2 from t_2 to t_3 must equal the heat required to heat W_1 from t_1 to t_3, so that

$$W_1(t_3 - t_1)(0.24 + 0.45H_1) = W_2(t_2 - t_3)(0.24 + 0.45H_2) \tag{10.30}$$

$$\therefore \frac{t_3 - t_1}{t_2 - t_3} = \frac{W_2(0.24 + 0.45H_2)}{W_1(0.24 + 0.45H_1)} \tag{10.31}$$

From equation 10.29

$$(H_3 - H_1)/(H_2 - H_3) = W_2/W_1 \tag{10.32}$$

Elimination of W_2/W_1 from equations 10.31 and 10.32 and solution for t_3 yields

$$t_3 = \frac{t_1 + t_2 \dfrac{H_3 - H_1}{H_2 - H_3} \cdot \dfrac{0.24 + 0.45H_2}{0.24 + 0.45H_1}}{1 + \dfrac{H_3 - H_1}{H_2 - H_3} \cdot \dfrac{0.24 + 0.45H_2}{0.24 + 0.45H_1}} \tag{10.33}$$

However, when $0.24 + 0.45H_2 \cong 0.24 + 0.45H_1$, equation 10.33 can be simplified to

$$t_3 = \frac{t_1H_2 - t_2H_1 + H_3(t_2 - t_1)}{H_2 - H_1} \tag{10.33a}$$

In equation 10.33*a*, t_3 is seen to be linear with H_3, as represented by the straight line of Fig. 10.6. The approximation is adequate for many engineering problems.

Example. Find the temperature that will result when air at 50°F with a humidity of 0.007 is mixed with enough air at 110°F and a humidity of 0.028 to give a humidity of 0.021 in the mixture. Equation 10.33*a* yields

$$t_3 = [50 \times 0.028 - 110 \times 0.007 + 0.021(110 - 50)]/(0.028 - 0.007) = 90°\text{F}$$

The precise solution from equation 10.33 is $t_3 = 90.505$°F.

10.15. Cooling, Dehumidifying. Where a stream of air at a temperature t_2 (Fig. 10.7) comes in contact with a heat-removing sink at a temperature, t_c, below the dew point, the state point of the cooled air falls on a straight line as shown in Fig. 10.7. The location of the final state point is dependent upon the heat-transfer characteristics of the medium separating the cooling material and the air.

This process is best represented by the cooling and dehumidification which results when air is passed through a finned-type unit refrigeration evaporator.* If we assume that a portion of the air is cooled to a temperature of t_c, the remainder staying at the t_2 condition, a process involving mixtures exists and the principles of the previous section apply.

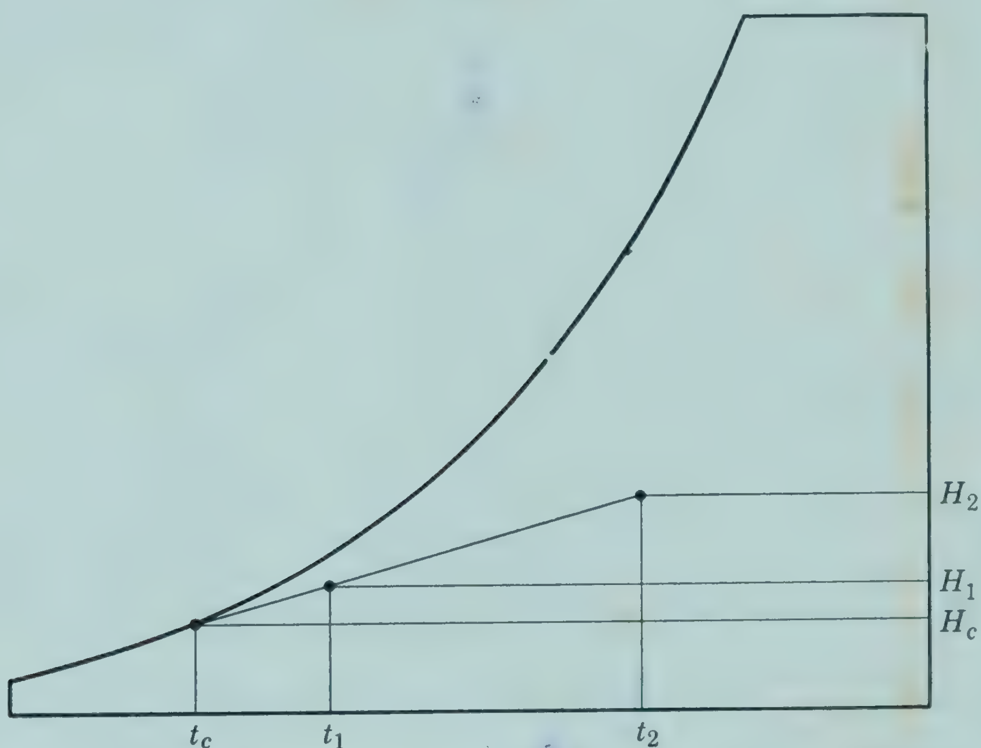


Fig. 10.7. The psychrometric process of cooling and dehumidifying.

The performance of a finned-type unit refrigeration evaporator is partially defined by the *by-pass factor, B.*

$$B = (t_1 - t_c)/(t_2 - t_c) = (H_1 - H_c)/(H_2 - H_c) \quad (10.34)$$

This factor designates that portion of the air assumed to by-pass the exchanger and subsequently to mix at t_2 and H_2 with the balance, which has been cooled and dehumidified to t_c and H_c.

10.16. Drying. Drying systems in which heat energy is supplied only by air, with sensible heat of the dry matter small in proportion to the latent heat of evaporation, and with negligible wall exchange, can be treated as cases of adiabatic humidification. As air passes over the material being dried, its temperature drops and its humidity rises, so that the wet bulb remains constant. This process is discussed in detail in Chap. 11.

* See Sect. 9.23 for the mathematical treatment of this type heat exchange.

10.17. Air Conditioning. Air conditioning involves processes of heating, cooling, humidifying, or dehumidifying, either singly or in suitable combination. Heating and cooling were treated in sect. 10.13.

Air may be humidified by passing it through a water spray or over saturated pads thus adding moisture to the air by an adiabatic process, *a-b*, Fig. 10.8. The cooled air is then heated to the desired temperature, *b-c*, Fig. 10.8.

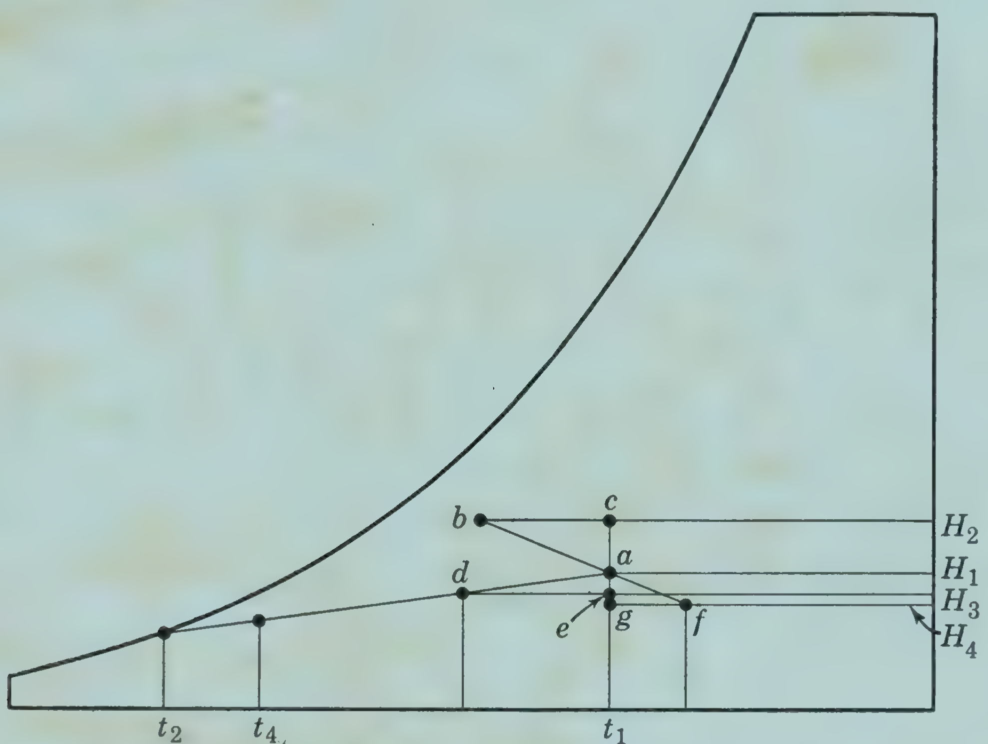


Fig. 10.8. The psychrometric air conditioning process.

Steam may be injected into the air thus increasing the humidity from *a* to *c* without essential change in temperature. This procedure is rigorous except for the sensible heat to be removed to cool the steam as vapor, from its introduction temperature to the air temperature. Precisely, the enthalpy of the mixture is the sum of the enthalpies of the components.

Dehumidification can be brought about by passing the air through a finned refrigeration cooler or a water spray at t_2 or brine spray at temperature t_4, cooling and dehumidifying from *a* to *d*, Fig. 10.8, then heating to *e*. Adsorbants such as silica-gel and activated alumina may be used to remove moisture by process *a-f*, then cooling.

10.18. A Psychrometric Chart. A psychrometric chart designed for the range 20° to 120°F is included as Fig. 10.9. This

chart can be used for any of the state or process determinations discussed in previous sections. Fig. 11.12 is a similar chart for a higher temperature range.

REFERENCES

1. Arnold, J. H. The Theory of the Psychrometer. *Physics* 4:255–262, 334–340. 1933.
2. Awbery, J. H., and Griffiths, Ezer. The Basic Law of the Wet and Dry-Bulb Hygrometer at Temperatures from 40C to 100C. *Proc. Phys. Soc. (London)* 44:132–142. 1932.
3. Carrier, W. H., and Mackey, C. O. A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems (PRO-59-7). *Trans. A.S.M.E.* 59:33–47 (Sect. 1). 1937.
4. Dropkin, David. The Deviation of the Actual Wet-Bulb Temperature from the Temperature of Adiabatic Saturation. *Cornell Univ. Eng. Expt. Sta. Bull.* 23. 1936.
5. Keenan, J. H., and Kaye, J. *Gas Tables.* Wiley. 1948.
6. Marvin, C. F. Psychrometric Tables for Obtaining the Vapor Pressure, Relative Humidity, and Temperature of the Dew Point from Readings of the Wet- and Dry-Bulb Thermometers. *U. S. Dept. of Comm. W.B.* 235. 1941.
7. Perry, John H. *Chemical Engineers Handbook.* Third Edition. McGraw-Hill. 1950.
8. Raber, B. F., and Hutchinson, F. W. *Refrigeration and Air Conditioning Engineering.* Pp. 148–202. Wiley. 1945.
9. Sherwood, T. K., and Comings, E. W. *Trans. Am. Inst. Chem. Engrs.* 28:88–117. 1932.
10. Wile, D. D. Psychrometry in the Frost Zone. *Refrig. Eng.* 48:291–303. 1944.
11. Zimmerman, O. T., and Lavine, Irvin. *Psychrometric Tables and Charts.* Industrial Research Service. 1945.

PROBLEMS

1. Dry- and wet-bulb temperatures are 80° and 65°F respectively. What are the humid volume, relative humidity, water-vapor pressure, dew point, absolute humidity?
2. Solve problem 1 relative to the top of a 10,000-ft mountain. The atmospheric pressure is 0.6876 that at sea level.
3. Atmospheric air has a temperature of 85°F and a dew point of 53°F. What is the relative humidity when cooled to 60°F? How many pounds of water are removed from 20,000 cu ft if cooled to 35°F?
4. A grain drier requires 12,000 cu ft per min of 115°F air. The atmospheric air is at 75°F and 68 per cent relative humidity. How many British thermal units per hour are required to heat the air?

5. Air at 95°F and 25 per cent relative humidity is blown through a 35°F water spray and is cooled to 75°F. What is the relative humidity?
6. What is the error in per cent resulting from chart design when the heat required to heat 65°F air with 90 per cent relative humidity to 105°F is determined from the adiabatic saturation lines?
7. The relative humidity on a 90°F day is 30 per cent.

   a. How many pounds of water vapor are in a room 20 ft by 30 ft by 10 ft high?
   b. What is the weight of the air-water-vapor mixture in the room above?
   c. Air is discharged from an evaporative cooler at 76°F. What is its relative humidity?
   d. Will eggs removed from a 45°F storage room "sweat"? Why?
   e. If the air is humidified isothermally, what is H at saturation?

8. A stream of air at 110°F and 10 per cent relative humidity merges with a stream at 70° and 90 per cent relative humidity. The temperature of the mixture is 76°F.

   a. What is the relative humidity of the mixture?
   b. If the higher temperature air stream has a rate of 9000 cu ft per min, what is the rate of the lower temperature stream?

9. Air at 80°F and 40 per cent relative humidity is cooled by a refrigeration coil which has a fin temperature of 33°F. The air is discharged from the coil at 48°F. The air rate is 6000 cu ft per min. How many pounds of water are removed from the air per 24 hr?

CHAPTER 11

Drying

NOMENCLATURE

A = area, sq ft.
c = a constant, dimensionless.
c_p = specific heat, Btu per (lb °F).
D_p = particle diameter, ft.
D_v = volumetric diffusivity, sq ft per hr.
D_{vw} = weight diffusivity of water vapor in air = $D_v\gamma/p$.
E = weight of moisture evaporated, lb.
e = natural logarithm base.
F = wet weight, lb.
f = surface thermal conductance, Btu per (hr sq ft °F).
f_v = surface water-vapor conductance, lb per (hr sq ft lb per sq in.).
G = weight velocity, lb per hr sq ft.
H_e = exhaust air humidity.
H_i = initial air humidity.
h = enthalpy of humid air, Btu per lb.
h_{fg} = latent heat of evaporation, Btu per lb.
k = thermal conductivity, Btu per (hr sq ft °F per ft).
M = moisture content, dry basis, per cent.
m = moisture content, wet basis, per cent.
M_E = equilibrium moisture content, dry basis, per cent.
M_f = final moisture content, dry basis, per cent.
M_i = initial moisture content, dry basis, per cent.
M_S = moisture content at surface, dry basis, per cent.
n = a constant, dimensionless.
P = dried weight, lb.
p = total pressure, lb per sq in.
p_a = pressure of water vapor in the air, lb per sq in.
p_s = water vapor pressure at saturation, t_s, lb per sq in.
Q = quantity of water, lb.
q = heat rate, Btu per hr.
rh = equilibrium relative humidity, a decimal.
S = surface conductance, water vapor, ft per hr.
T = absolute temperature, °R.

t_a = air temperature, °F.
t_s = water-surface temperature, °F.
V = air rate, cu ft per (min sq ft).
V_h = velocity, ft per hr.
v = humid volume, cu ft per lb dry air.
W_d = weight of dry material, lb.
W_m = weight of moisture, lb.
w = water removal rate, lb per min.
x = distance from center of mass, ft.
x_h = fictive film thickness offering resistance to heat flow, ft.
x_v = fictive film thickness offering resistance to diffusion, ft.
α = generalized drying index, inverse hr.
α' = specific drying index.
γ = specific weight, lb per cu ft.
μ = viscosity, lb per hr ft.
θ = time, hr.

The removal of moisture from a product is known as drying or dehydration. Although these terms are used interchangeably, *drying* is the removal of moisture to a moisture content in equilibrium with normal atmospheric air or to such a moisture content that decrease in quality from molds, enzymic action, and insects will be negligible—12 to 14 per cent wet basis—for most materials. *Dehydration* is the removal of moisture to a very low moisture content, nearly bone-dry condition. Bone-dry material is material from which all the moisture has been removed; the moisture content is zero.

The importance of drying farm products is increasing. Drying permits the farmer to secure a greater economic return for the following reasons:

1. Early harvest (at high moisture content) minimizes field damage and shatter loss and facilitates tillage operations for such products as corn, small grains, and grass seed.
2. Long-period storage without product deterioration is possible.
3. Viability of seeds is maintained over long periods.
4. Production operations are facilitated for such products as cotton and corn.
5. Products with greater economic value are produced, for example tobacco, dried fruit, and vegetables.
6. Waste products can be converted to useful products, for example livestock feed from fruit pulp and almond hulls.

MOISTURE CONTENT

The moisture content of a substance is usually expressed in percentage by weight on the wet basis, i.e., in grams of moisture per 100 g of sample.

$$m = 100W_m/(W_m + W_d) \tag{11.1}$$

where m is moisture content, wet basis, per cent; W_m, weight of moisture; W_d, weight of bone-dry material.

This method of expression tends to give an incorrect impression when applied to drying, since both the moisture content and the basis on which it is computed change as drying proceeds. If, however, the moisture is expressed as moisture content, dry basis, parts of water per part of "bone-dry" matter (water-free matter), a correct representation of moisture to be removed and of drying rate can be obtained, since the amount of dry matter remains constant as the moisture evaporates.

Moisture content, dry basis, per cent, M, is

$$M = 100(W_m/W_d) = 100m/(100 - m) \tag{11.2}$$

The moisture content, dry basis, is sometimes expressed as moisture ratio, that is, pounds of moisture per pound of dry matter or $M/100$. The quantity of moisture present at any time is directly proportional to the dry-basis moisture content.

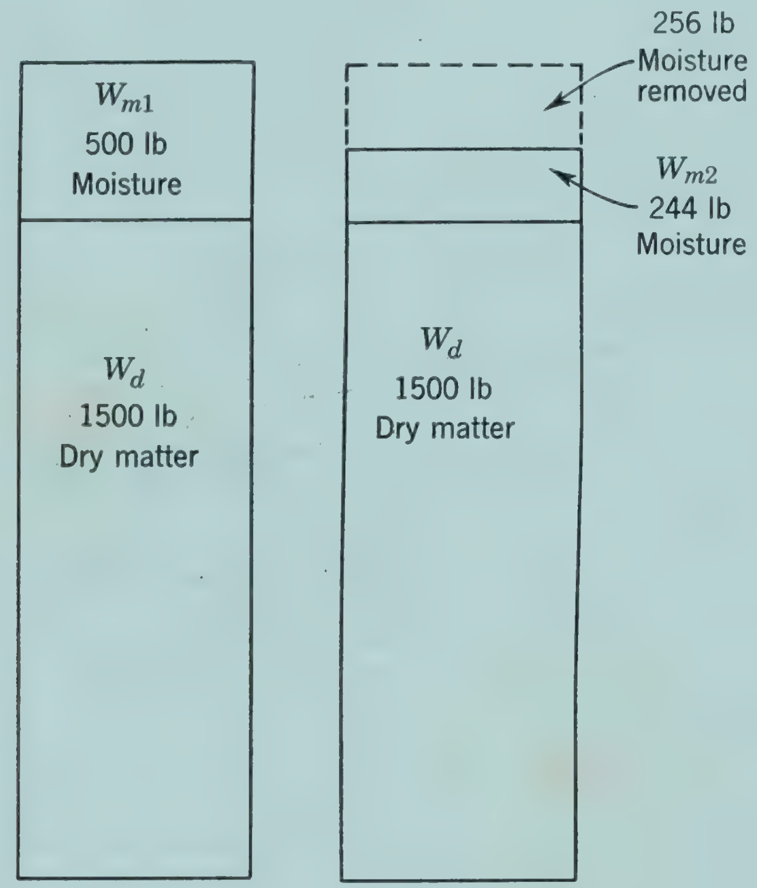
Example. Find the amount of moisture to be removed in drying a ton of grain, initially at 25 per cent moisture (wet basis), to 14 per cent moisture (wet basis). Also find the weight of dried grain.

Solution (*a*), from wet-basis moisture contents. The initial quantity of moisture is $(25/100) \times 2000$ or 500 lb. The dry matter is 2000 to 500 or 1500 lb. The final weight of 14 per cent moisture grain is the weight of dry matter, multiplied by the pounds of moist material per pound of dry matter, or $1500 \times 100/(100 - 14) = 1744$ lb. Thus the amount of moisture to be removed is $2000 - 1744$ or 256 lb.

Solution (*b*), from dry-basis moisture contents. The initial moisture content, dry basis, is $100 \times 500/1500$ or 33.33 per cent. This can also be found as $25/(100 - 25)$. The final moisture content, dry basis, is $100/(100 - 14)$ or 16.28 per cent. The amount of moisture to be removed, per ton of wet grain, is then $1500\ (33.33 - 16.28)/100$ or 256 lb. The yield is $2000 - 256$ or 1744 lb. The amount of dry matter is $2000 \times 100/(100 + 33.33) = 1500$ lb. *Note:* The amount of moisture to be removed is not the initial weight

multiplied by the difference in wet-basis moisture contents. It is equal to the weight of dry matter, multiplied by the difference in dry-basis moisture contents.

When only initial and terminal values are needed, the moisture content, wet basis, can, with appropriate equations, be as readily used as the moisture content, dry basis, as shown by the following.



F = wet weight = $W_m + W_d$, lb.
E = weight of moisture evaporated, lb.
P = dried weight, lb.
W_d = weight of dry matter, lb.

Wet Basis from m_1 to m_2		*Dry Basis from M_1 to M_2*	

Pounds of moisture to be removed per pound of fresh material

$$\frac{E}{F} = \frac{m_1 - m_2}{100 - m_2} \quad (11.3) \qquad\qquad \frac{E}{F} = \frac{M_1 - M_2}{M_1 + 100} \quad (11.4)$$

Pounds of moisture to be removed per pound of dried product

$$\frac{E}{P} = \frac{m_1 - m_2}{100 - m_1} \quad 11.5) \qquad\qquad \frac{E}{P} = \frac{M_1 - M_2}{M_2 + 100} \quad (11.6)$$

Wet Basis from m_1 to m_2 | *Dry Basis from m_1 to m_2*

Pounds of wet material to produce a pound of dried product

$$\frac{F}{P} = \frac{100 - m_2}{100 - m_1} \tag{11.7}$$

$$\frac{F}{P} = \frac{M_1 + 100}{M_2 + 100} \tag{11.8}$$

Pounds of dried product per pound of wet material

$$\frac{P}{F} = \frac{100 - m_1}{100 - m_2} \tag{11.9}$$

$$\frac{P}{F} = \frac{M_2 + 100}{M_1 + 100} \tag{11.10}$$

Pounds of moisture to be removed per pound of dry matter

$$\frac{E}{W_d} = \frac{100(m_1 - m_2)}{(100 - m_1)(100 - m_2)} \tag{11.11}$$

$$\frac{E}{W_d} = M_1 - M_2 \tag{11.12}$$

MOISTURE DETERMINATIONS

Moisture-determining procedures are classed as primary or direct and secondary or indirect. The primary procedures are such that the moisture in a sample is removed and the quantity is determined by weighing or measuring. The secondary procedures depend upon some characteristic of the material which is related to moisture content and must be calibrated against an official primary method.

"Official" methods are those that have been accepted by the Association of Official Agricultural Chemists and are recognized by the U. S. Production and Marketing Administration. Moistures determined by "Official" methods by certified inspectors are accepted by the courts.

A list of methods and some commercial moisture-determination devices follow:

11.1. Primary Methods, Oven ("Official"). Samples are ground and dried in an air or vacuum oven at a temperature close to boiling water for a prescribed time. The loss in sample weight is considered to be moisture. Temperature and drying time vary from material to material. The specific procedure for each material must be followed.

Operating conditions for a few materials are listed as follows. Temperatures are Fahrenheit.

Grain and stock feeds: 204° to 212°, 5 hr, vacuum oven; or 271° to 279°, 2 hr, air oven.

Dried fruits: 158°, 6 hr, vacuum oven. (Also tentative for dried vegetables.)

Hops: 140°, 3 hr, vacuum oven; or 218°, 1 hr, air oven.

Nuts (tentative): 158° vacuum oven, weight at 2-hr intervals until loss per interval does not vary more than 3 mg per 2-g sample.

Dried milk powder: 202°, 2 hr, vacuum oven.

Molasses: 140°, 2 hr, vacuum oven.

Toluene distillation ("Official"): the ground sample is distilled in toluene, 232°, until all the water has been removed from the sample, about 1 hr. Xylene ("unofficial") which boils at 280° is also used in this system. Both are a fire hazard.

Brown-Duvel Moisture Tester (accepted as "Official"): whole grains are distilled in a mineral oil that has a higher boiling point than the distilling temperature. The moisture that is driven off is condensed and measured.

11.2. Secondary Methods. *Electrical Resistance Meters.* Devices that measure the electrical resistance of products are calibrated against oven determinations and are adequate for many tests. Since resistance varies with the distribution of moisture within the material, with material density, and perhaps with acid index and other factors and since the characteristics of the moisture machine itself change with time, exact results cannot be expected. Hay samples that contain a few wet pieces may give a completely erroneous indication of moisture content. Material removed from a drier for a moisture check has a moisture gradient through each element and may also yield unsatisfactory results. Some studies with wheat[5] indicate errors of a quarter of a per cent can be expected. Errors greater than this are frequently experienced. Tests can be made in a minute or less.

Dielectric Meters. The capacitance of an electrical condenser varies with the moisture content of a material placed between its plates. This feature is used in dielectric meters for moisture determination. Meters of this type are as fast as the resistance meters but are less accurate.[5]

EQUILIBRIUM MOISTURE CONTENT

Farm products, both natural and processed, contain adsorbed moisture. The adsorbed moisture exerts a moisture vapor pres-

sure which varies with the moisture content of the material and from material to material. The ratio of the moisture vapor pressure to the saturated vapor pressure of pure water at the temperature of the material is called the equilibrium relative humidity. A plot of the equilibrium relative humidity (abscissa) and moisture content (ordinate) is known as an equilibrium moisture curve.

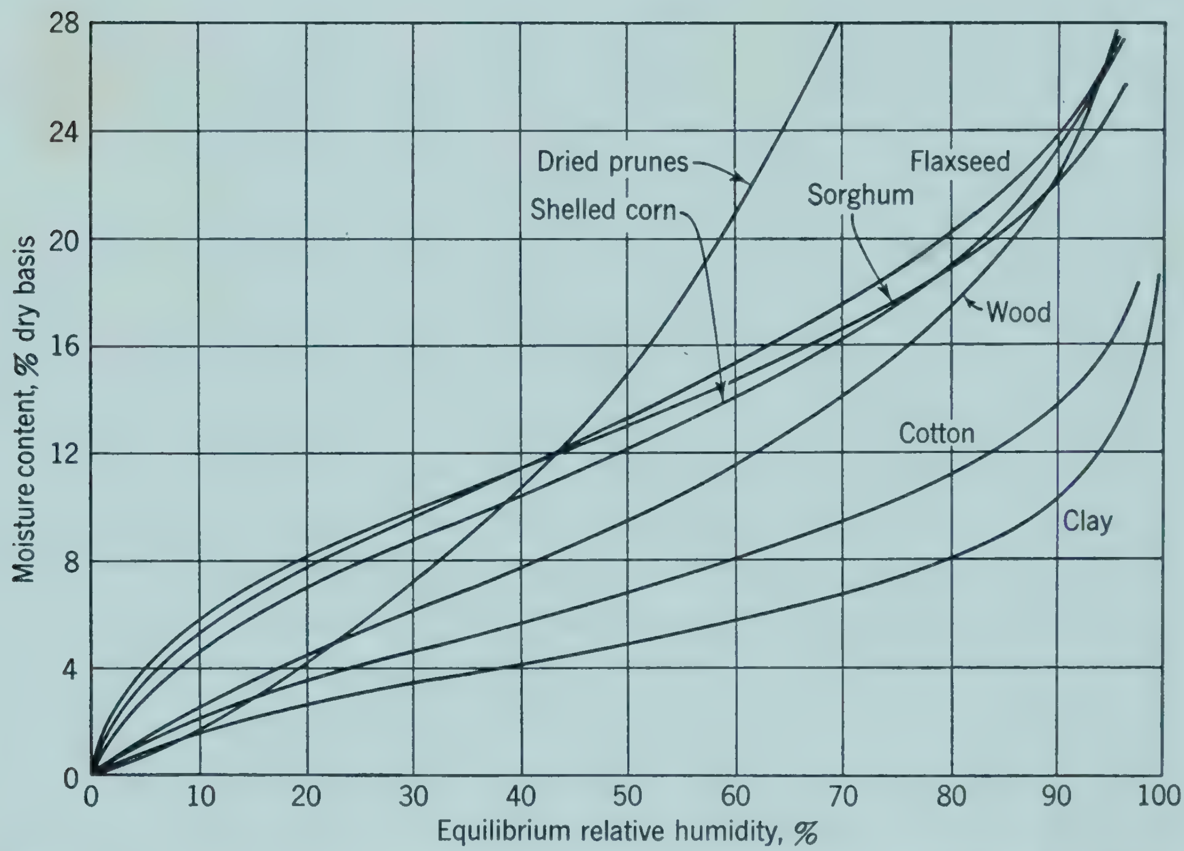


Fig. 11.1. Equilibrium moisture curves of a number of materials at room temperature, approximately 77°F.

Such a plot for a number of materials is given in Fig. 11.1. Materials that exhibit this characteristic are called hygroscopic materials.

The moisture content may be given on either a wet or dry basis, depending upon the intended use of the curves. The curves are affected somewhat by a change in temperature, an increase in temperature shifting the curve downward. The effect is not sufficiently pronounced to be considered in most engineering work.

Equilibrium moisture curves can be defined by the following equation [4]

$$1 - rh = e^{-cT}{M_E}^n \tag{11.13}$$

where rh = equilibrium relative humidity, a decimal.
M_E = equilibrium moisture content, dry basis, per cent.
T = temperature, °R.
c, n = constants.

Equilibrium moisture data can, therefore, be reported in terms of the constants c and n. Values of these constants for some materials are given in Table 11.1.

Table 11.1 VALUES OF EQUILIBRIUM CONSTANTS c AND n FOR SOME MATERIALS

Material	c	n
Shelled corn	1.10×10^{-5}	1.90
Wheat	5.59×10^{-7}	3.03
Sorghum	3.40×10^{-6}	2.31
Soybeans	3.20×10^{-5}	1.52
Flaxseed	6.89×10^{-6}	2.02
Raisins	7.13×10^{-5}	1.02
Dried peaches	4.11×10^{-4}	0.564
Dried prunes	1.25×10^{-4}	0.865
Cotton	4.91×10^{-5}	1.70
Wood	5.34×10^{-5}	1.41
Spray-dried eggs	2.95×10^{-5}	2.00
Natural clay	7.53×10^{-5}	1.72

The equilibrium moisture properties of materials are important in storage and drying. If the relative humidity of the air in contact with a material is higher than the equilibrium relative humidity of the material at its current moisture content, the material will increase in moisture content, the moisture content at the air relative humidity being the value approached. An air relative humidity lower than the equilibrium will cause the moisture content to decrease.

DRYING PROCESSES

Drying processes can be divided into two periods: (1) the constant drying-rate period, and (2) the falling drying-rate period.

11.3. Constant-Rate Period. In this period a material or mass of material containing so much water that liquid surfaces exist will dry in a manner comparable to an open-faced body of water. The water and its surroundings, not the solid, will deter-

mine the rate of drying. Wet sand, soil, pigments, and washed seed are examples of materials that dry at a constant rate at first.

The heat energy for drying can be applied as radiant heat energy, e.g., from infrared lamps; as conducted heat, e.g., tumbling the material in contact with the walls of a hot drum; and as convected heat from hot air. The last is represented by the adiabatic evaporation equation as from a wet bulb, sect. 10.9. If drying is by passing of air through the mass, the following heat and mass balance exists:

$$\frac{dW}{d\theta} = f_v A(p_s - p_a) = \frac{fA(t_a - t_s)}{h_{fg}} \tag{11.14}$$

where $dW/d\theta$ = drying rate, lb water per hr.

f = thermal conductance of the air film at the water-air interface, Btu per (hr sq ft °F).

A = water surface area, sq ft.

h_{fg} = latent heat of water at t_s, Btu per lb.

t_a = air temperature, °F.

t_s = water surface temperature, wet-bulb temperature, °F.

f_v = water-vapor transfer coefficient at the water-air interface, lb per (hr sq ft lb per sq in.).

p_s = water-vapor pressure at t_s, lb per sq in.

p_a = water-vapor pressure in the air, lb per sq in.

Values of f_v and f were determined by Gamson, Thodos, and Hougen,[2] for drying by forcing air through beds of moist spherical or cylindrical pellets. In the transfer of vapor from the surface of the solid to the moving air stream, the vapor must first pass through a laminar layer of moist air and then usually through an adjacent turbulent zone. The change in partial pressure of vapor with distance from the surface is shown schematically in Fig. 11.2. The vapor conductance f_v can be regarded as the diffusivity of vapor through air, D_{vw}, divided by the thickness, x_v, of a fictive layer which vapor passes through only by diffusion. This is analogous to heat transfer, where the surface thermal conductance $f = k/x_h$.

For vapor transfer, with Reynolds number greater than 350,

$$\frac{D_p}{x_v} = \frac{f_v D_p}{D_{vw}} = 0.989 \left(\frac{D_p G}{\mu}\right)^{0.59} \left(\frac{\mu}{D_{vw} p}\right)^{1/3} \tag{11.15}$$

and for heat transfer,

$$\frac{D_p}{x_h} = \frac{fD_p}{k} = 1.064\left(\frac{D_pG}{\mu}\right)^{0.59}\left(\frac{c_p\mu}{k}\right)^{1/3} \tag{11.16}$$

where D_p = diameter of particle, ft (diameter of equivalent sphere for nonspherical particles).

D_{vw} = weight diffusivity of water vapor in air. (Note: $D_{vw} = D_v\gamma/p$.)

D_v = volumetric diffusivity, sq ft per hr.

x_v = fictive film thickness offering resistance to diffusion, ft.

x_h = fictive film thickness offering resistance to heat transfer, ft.

G = weight velocity, lb per hr sq ft (Note $G = V_h\gamma$).

V_h = velocity, ft per hr.

γ = specific weight, lb per cu ft.

μ = viscosity, lb per hr ft.

c_p = specific heat, Btu per lb, °F.

p = total pressure, lb per sq in.

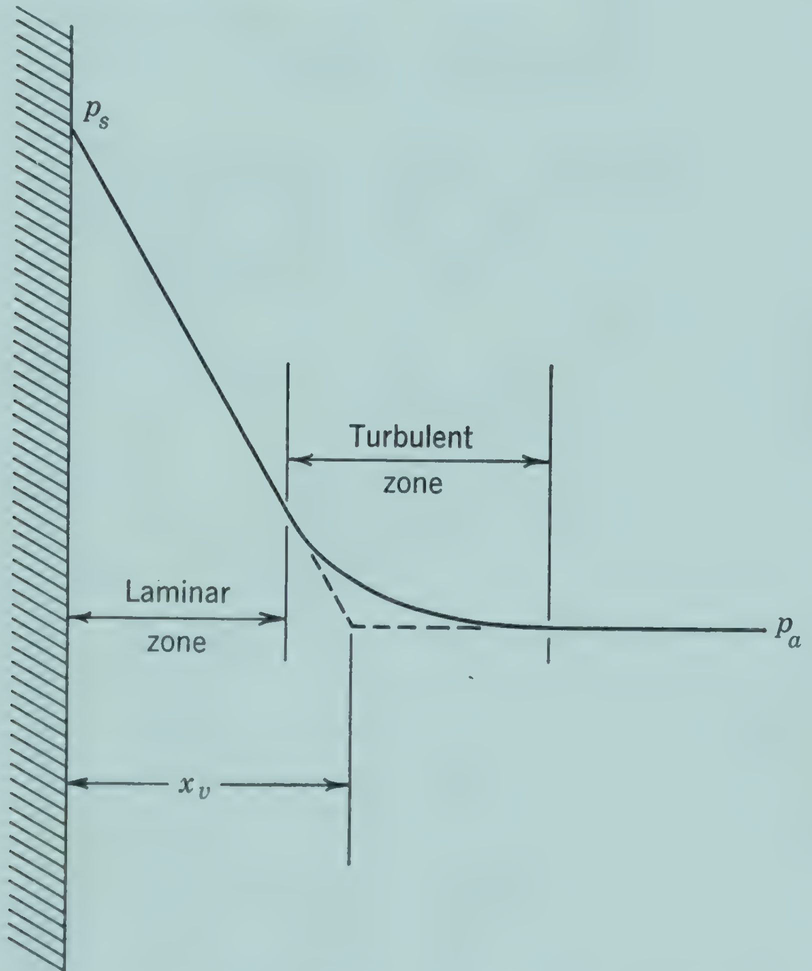


Fig. 11.2. Change of partial pressure of water vapor with distance from surface for a constant drying-rate condition.

The constant-rate drying period will proceed until free moisture disappears from the surface; the moisture removal rate will then become progressively less. The moisture content at which the drying rate ceases to be constant is known as the critical moisture content. No effective open moisture surfaces exist below the critical moisture content, and the drying mechanism of the constant-rate period no longer holds.

11.4. Falling-Rate Period. Practically all agricultural drying takes place in the falling-rate period. Products that are moved into a drier from a washer may experience a short initial constant-rate period. This period is usually minor when compared to the complete drying process and can be neglected in the calculations.

The falling-rate period is bounded by equilibrium moisture contents of an equilibrium moisture curve between zero and nearly 100 per cent relative humidity. Moisture contents near the 100 per cent level would be in the constant-rate period. The 100 per cent equilibrium point is not a satisfactory procedure for determining the exact critical moisture content since equilibrium moisture data observed above 95 per cent relative humidity are usually unreliable.

11.5. The Falling-Rate Drying Mechanism. Drying in the falling-rate period involves two processes, (1) movement of moisture within the material to the surface and (2) removal of the moisture from the surface.

The internal moisture-movement mechanism has been studied by a number of investigators.[6,8,14,15,17,19,21]

Finely divided materials containing cell or void cavities and which are essentially not hygroscopic permit internal moisture movement by capillary and gravitational flow. Moisture above the saturation point in leather, cotton, paper, etc., and above the equilibrium moisture range in finely divided materials such as soil, ceramic stock, paint, pigments, etc., moves by this mechanism. Hygroscopic and non-hygroscopic materials dry comparably.

Moisture movement is by *liquid diffusion* if the moisture content is below the saturation point or within the equilibrium moisture range or if the material is a single-phase material such as soaps, glues, and pastes. The movement is analogous to that of heat conduction in a solid, and the following equations apply:

Within the solid

$$(1/A)(\partial Q/\partial \theta) = -D_v\gamma(\partial M/\partial x)/100 \qquad (11.17)$$

At the surface

$$-D_v\gamma(\partial M/\partial x) = S(M_S - M_E)\gamma \qquad (11.18)$$

$$\partial M/\partial \theta = D_v(\partial^2 M/\partial x^2) \qquad (11.19)$$

Q = quantity of water, lb.
D_v = diffusivity, sq ft per hr.
γ = dry-solid density, lb per cu ft.
M = moisture content, dry basis, per cent.
M_S = moisture content at surface, dry basis, per cent.
M_E = equilibrium moisture content at the relative humidity of the drying air, dry basis, per cent.
S = surface conductance, ft per hr.
x = distance from center of the mass being dried, ft.

The applicability of these relationships has been studied by a number of investigators.

Solution of these equations was given by Newman [8] for a material of constant diffusivity which dries without shrinking in surroundings that provide a constant atmosphere and constant surface conductance and leads to curves similar to those for heat conduction in solids (Figs. 9.2, 9.3, 9.4).

Observations on large-size objects such as lumber have shown that the observed moisture gradients through the unit differ from those calculated from a constant diffusivity. The lack of coincidence is due to the fact that the diffusion coefficients vary with moisture content, temperature, pressure, and material density, all of which usually vary during drying.

The mechanism of moisture removal at the surface as set out by Newman [8] is shown in equation 11.18. The moisture is assumed to be brought to the surface by diffusion and evaporated to a vapor at the surface. The vapor is then removed from the surface by conduction through the air film to the moving air. The driving force is the difference in moisture content at the surface and the equilibrium moisture content of the material at the air state.

Experimental drying studies of agricultural products have shown that the drying rate is proportional to the difference in

moisture content between the material being dried and the equilibrium moisture content at the drying air state or:

$$dM/d\theta = -\alpha'(M - M_E) \tag{11.20}$$

Equation 11.20 is analogous to Newton's law of cooling. It could be expected to hold quite well where the diffusivity of moisture within the solid is high with respect to the surface conductance and thickness. This is not true for grains, fruits, and vegetables, however, where a substantial moisture gradient exists within the material during drying. The application to fruits and grains is made more plausible by the artifice of replacing the distributed internal resistance by a single lumped resistance at the surface, in series with the surface vapor resistance. The equilibrium moisture content that must be postulated to secure a linear plot of M against θ, time, on semilogarithmic coordinates is often found considerably above the equilibrium values from hygrostatic measurements at constant weight.

Additional limited studies have indicated that air velocity and temperature are probably related to the drying rate, thus:

$$\alpha' = \alpha V^n p_s \tag{11.21}$$

where V = air rate, cu ft per (min sq ft).
p_s = saturated water vapor pressure at the temperature of the drying air, lb per sq in.

The velocity exponent n is an indication of the relative effect of internal diffusion as compared to surface resistance upon the drying rate. If n is 0.6, there is no internal resistance to moisture movement, and resistance to vapor transfer at the surface controls the drying rate. Small values of n indicate that the internal resistance to flow controls the drying rate and that the surface resistance is minor.

The saturated-pressure factor p_s is included in the drying index because the removal of moisture from the surface is a vapor-pressure mechanism, thus the driving force is proportional to the saturated pressure. The internal driving force may be considered as vapor pressure in which case $(M - M_E)$ of equation 11.20 may be replaced with $(rh - rh_E)$, the respective equilibrium relative humidity values. This expression can be represented by $(1/p_s)(p - p_E)$ in which p_E is the equilibrium vapor pressure

exerted by the material. This rational concept of the driving force for drying is additional evidence supporting the presence of the p_s term in equation 11.21.

Equations 11.20 and 11.21 when combined and integrated for constant temperature, humidity, and velocity with limits give

$$(M_\theta - M_E)/(M_0 - M_E) = e^{-\alpha V^n p_s \theta} \qquad (11.22)$$

M_0 is the initial average moisture content at zero time; M_θ, the average moisture content after a period of time θ. The time θ is usually expressed in hours.*

11.6. Heat and Mass Balance. Drying with heated air is an adiabatic process, the energy for moisture evaporation being supplied by a reduction in temperature of the air. The wet-bulb (adiabatic humidification) lines of the psychrometric chart and equation 11.20 represent this process and can be used for calculating a drying heat and mass balance.

Air of state a, Fig. 11.3, is heated to state b and passed through the material to be dried. The state point moves up the wet-bulb line, and the air exhausts at state d. The water transport rate is

$$w = \frac{V}{v}(H_d - H_b) \qquad (11.23)$$

where w = water removable rate, lb per min.

V = air rate, cu ft per min per sq ft.

v = humid volume of air at point of rate measurement, cu ft per lb dry air.

The moisture removal rate from the drying-rate equation is

$$w = (W_d/6000)\alpha V^n p_s (M - M_E) \qquad (11.24)$$

W_d is the pounds of material (bone dry) per square foot through which the air is passing. Therefore, the change in humidity is

$$\Delta H = H_d - H_b = \frac{W_d \alpha p_s v}{6000 V^{1-n} A}(M - M_E) \qquad (11.25)$$

* Even though the above-mentioned drying-rate procedure is satisfactory for agricultural materials, the student should realize that future rigorous studies may develop drying principles that will be more applicable generally and some of the currently acceptable approximations may become obsolete.

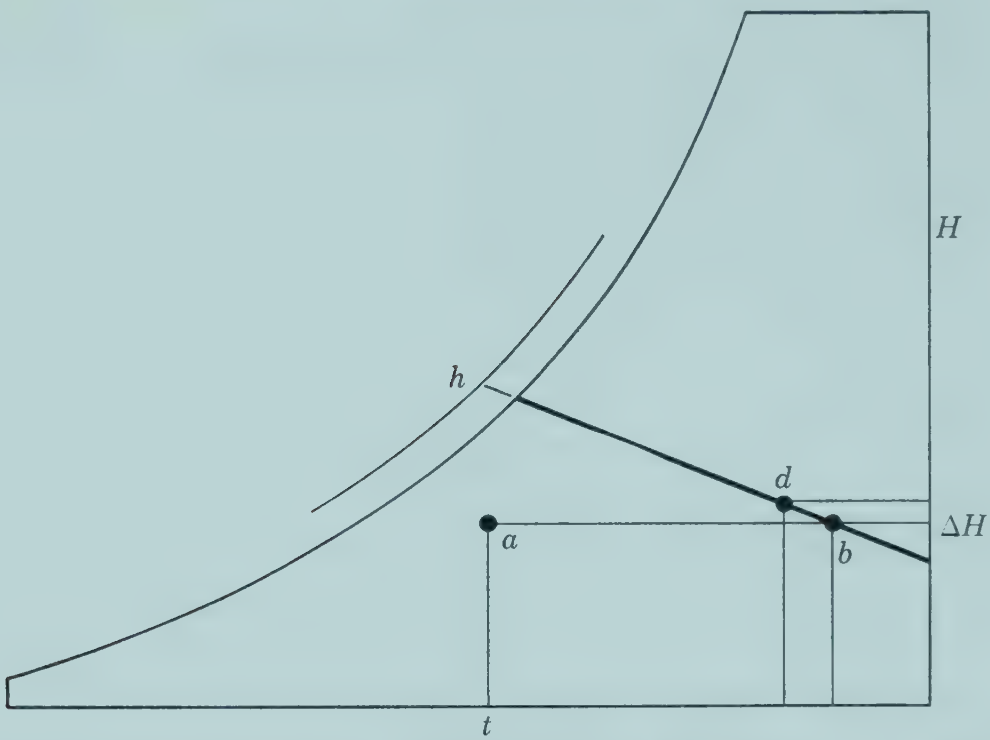


Fig. 11.3. The air state change, b to d, for a drying process. See also Fig. 11.12.

The energy that must be supplied to heat the air can be found with the help of the psychrometric chart and is

$$q = \frac{60VA}{v}(h_b - h_a) = \frac{60VA}{v}(t_b - t_a)(0.24 + 0.45H_a)$$

q is the heat rate in British thermal units per hour for the air rate represented by V.

The thermal efficiency in per cent may be expressed as:

$$\frac{(dM/d\theta)W_d h_{fg}}{q} \tag{11.26}$$

In this definition of thermal efficiency, which is commonly used for drying with heated air, the system is charged with the heat energy of the fuel and is credited with the latent heat of evaporation. The latent heat h_{fg} is at the exit temperature of the drying air.

An equivalent definition is the ratio of the air temperature drop in the drier to the rise in the heater. In drying with unheated air, these definitions indicate an infinite efficiency; with mild heating, efficiencies above 100 per cent may occur.

11.7. Limitations of the Drying Equations. The drying equations discussed in sect. 11.5 are based on the thin-layer drying concept. The "thin layer" dries uniformly and at state conditions defined by Fig. 11.3. No gradients are assumed. This concept does not hold rigorously, however, for the following reasons:

1. The temperature, humidity, saturated-vapor pressure, and specific volume of the air change from state b to state d (Fig.

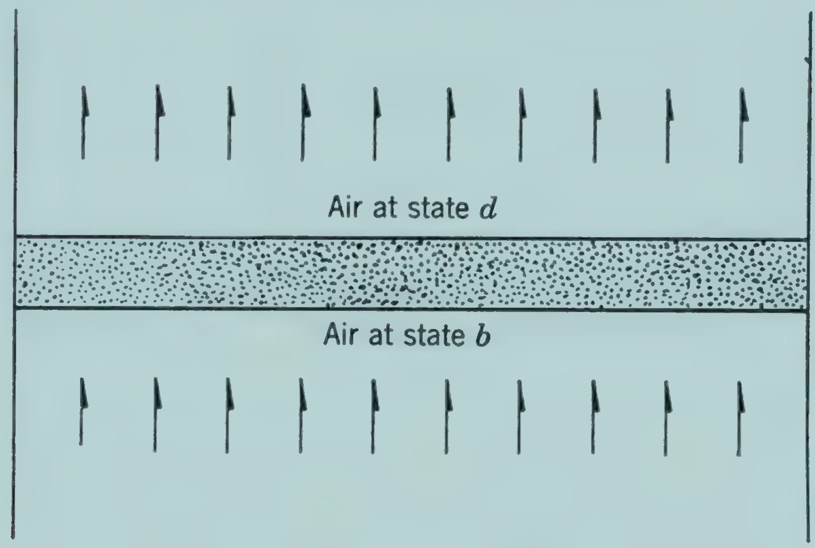


Fig. 11.4. Drying in a "thin" layer.

11.3) as it passes through the layer. Consequently, the drying potential will decrease because of a decrease in p_s and an increase in relative humidity which determines M_E.

2. Owing to the variables listed in (1) above, the top of the layer, Fig. 11.4, will dry at a slower rate and will have a higher moisture content after a period of time than the lower portion of the layer. The difference in moisture content between the top and bottom will increase progressively to a maximum and then decrease as drying progresses.

3. The calculations do not recognize the heat required to adjust the temperature of the grain initially or as drying progresses. This quantity is usually small as compared to the latent heat and usually may be neglected. Heat loss from the walls by radiation and conduction may be significant.

4. At low moisture contents, the energy required to release the moisture becomes greater than the latent heat by the heat of wetting. This factor is estimated to become significant below

moisture contents of 5 or 6 per cent, variation existing from material to material.

11.8. Approximate Deep-Bed Calculations. A "deep bed" is contrasted to a "thin layer" by recognizing a significant air-travel distance through the material, the effect being nonuniform drying. An approximate solution can be made by a stepwise integration which is demonstrated by the following example:

A 6-in. layer of shelled corn is to be dried from 20 per cent moisture content (wet basis) to 14 per cent with 120°F air at 16 per cent relative humidity. The air rate is 25 cu ft per min per sq ft. The assumed drying coefficients are $k = 0.23$, $n = 0.1$.

Consider six 1-in. layers with drying elements of 15 min. Assume thin-layer drying and prepare a table of calculations as shown in the accompanying tabulation. Each layer increment is assumed to dry uniformly with equation 11.22 applying. H_1, RH_1, t_1, p_s, and M_E are determined from the state of the air entering the layer

Depth, ft		Hr 0–0.25	Hr 0.25–0.5	Hr 0.5–0.75	Hr 0.75–1.00	Hr 1.00
0.167–0.25	t_2					
	RH_2					
	H_2					
	M	25.0	24.1	23.2	22.3	
	M_E	11.5	11.0	10.5		
	p_s	0.841	0.894	0.949		
	t_1	96	98	100		
	RH_1	47	$42\frac{1}{2}$	39		
	H_1	0.0170	0.0165	0.0161		
0.083–0.167	t_2	95	97	99	101	
	RH_2	50	44	41	37	
	H_2	0.0174	0.0168	0.0163	0.0158	
	M	25.0	23.5	22.0	20.6	
	M_E	8.3	8.0	7.7	7.5	
	p_s	1.168	1.238	1.312	1.312	
	t_1	107	109	111	111	
	RH_1	28	25	$23\frac{1}{2}$	$22\frac{1}{2}$	
	H_1	0.0144	0.0141	0.0138	0.0135	
0–0.083	t_2	106	108	$110\frac{1}{2}$	111	112
	RH_2	30	26	24	23	22
	H_2	0.0146	0.0142	0.0139	0.0136	0.0133
	M	25.0	22.6	20.5	18.7	17.1
	M_E	6.0	6.0	6.0	6.0	
	p_s	1.692	1.692	1.692	1.692	
	t_1	120	120	120	120	
	RH_1	16	16	16	16	
	H_1	0.0114	0.0114	0.0114	0.0114	

increment and represent the average for the time increment shown. M is the average initial moisture content for the layer at the time shown. H_2, RH_2, and t_2 are state conditions for the air leaving the layer at the time shown. The air entering the grain has a temperature of 120°F and a relative humidity of 16 per cent. These values establish the psychrometric adiabatic humidification line upon which the air state moves as drying progresses.

The calculated average moisture content of the bottom layer after, for example, ½ hr of drying is 20.5 per cent. The characteristics of the air leaving the bottom layer and entering the second layer are found by calculating the humidity increase in the air by equation 11.25. The increase added to 0.0114 is 0.0142. The later value H_2 is applied to the psychrometric adiabatic line from which values of RH_2 and t_2 are taken. H_1, RH_1, and t_1, for the second layer, are respective averages of the subscript-2 values for the air leaving the lower layer at the beginning and end of the period. The saturated pressure p_s, 1.238, is that at t_1. M_E is taken from Fig. 11.1 for RH_1, or 8 per cent. The calculations for the second layer progress from left to right, p_s and RH_1 changing for each time calculation.

The above procedure can be extended both as to depth and time until the desired operating range is covered. The errors discussed previously can be minimized by using small increments of time and depth.*

DRYING PROCEDURES

Agricultural materials must be dried by different procedures because of their inherent characteristics which may be classified on the basis of the following factors:

1. *Temperature Tolerance.* High temperatures may reduce germination, partially cook the product, or change its chemical or physical characteristics.
2. *Humidity Response.* Materials that undergo physiological or other change during drying, e.g., tobacco, lumber, prunes, may have to be dried with air of a specific moisture content.

* The errors can be further minimized by an appropriate numerical procedure such as the first modification of Euler's method. See J. B. Scarborough, *Numerical Mathematical Analysis,* The Johns Hopkins Press, 1930.

3. *Compression Strength.* Materials that crush or deform under pressure, e.g., fruit and vegetables, must be dried in thin layers; ear corn can be dried in deep beds, tobacco must be suspended.

4. *Fluidity.* Loose hay, ear corn, and other poor-flowing materials cannot be dried in a continuous-flow drier. The angle of repose (sect. 2.23) affects drier type and design.

The procedure and type of equipment recommended for a specific installation will depend upon the factors listed above, the quantity to be dried and drying rate required, weather conditions, and various economic factors.

11.9. Batch or Bin Driers. The material to be dried is placed in a bin or container, and air is forced through the mass until

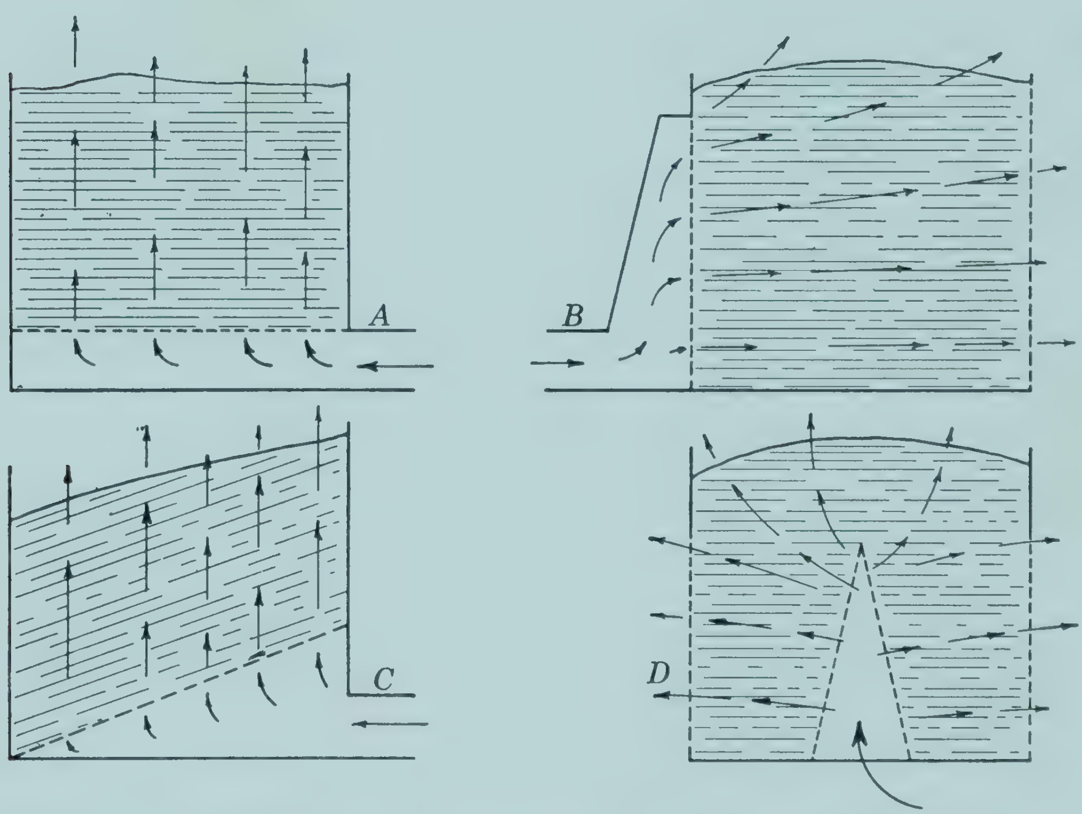


Fig. 11.5. Some deep-bed drying arrangements.

dry. Arrangements such as shown in Fig. 11.5 are frequently used.

The systems are simple, moderately inexpensive, and serve as storage units after drying is completed. Labor requirements are high since the bins are not entirely self-emptying. Design *C*, Fig. 11.5, utilizes the angle of repose of the grain and is nearly or completely self-emptying. Air distribution in designs *B* and

D is not uniform. Consequently, nonsymmetrical drying may result.

Materials to be dried by this system must have sufficient compressive resistance to resist crushing under load and to maintain the normal void space so that proper air rates can be maintained. Resistance to air flow limits the depth for highly resistant materials since adequate air rates are possible only with excessively large power units.

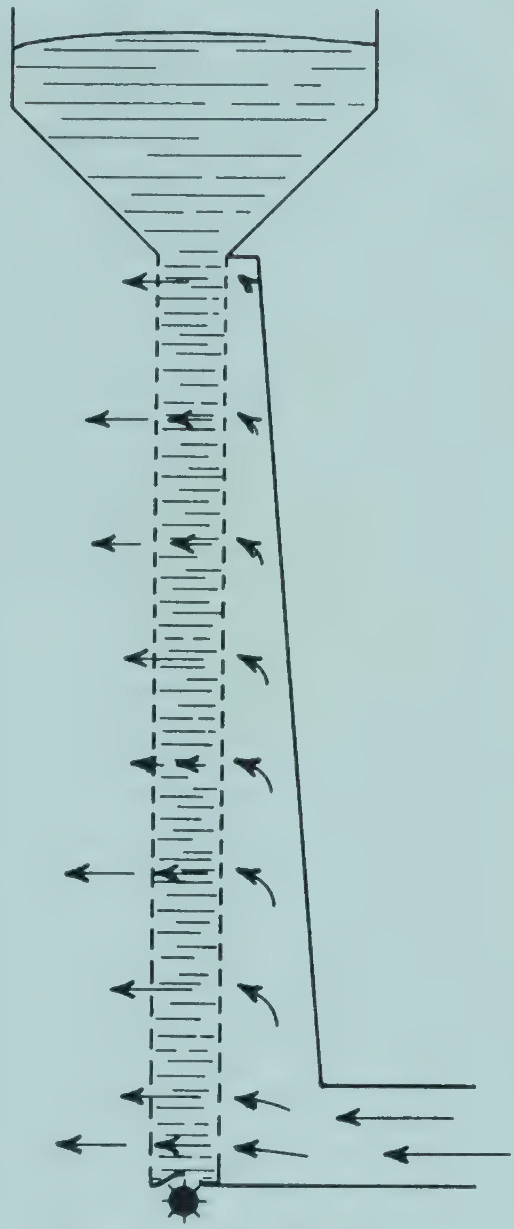


Fig. 11.6. The column drier.

The mass dries progressively in the direction of air flow. The part of the mass in the air discharge region is subject to high humidities and moderate temperatures and may spoil from mold before the moving drying front has reached it. Adequate drying of the mass in the air discharge region is accompanied by overdrying of the mass in the air entrance region. This undesirable feature can be minimized by (1) drying with the lowest practicable air temperature, (2) using the highest practicable air rate, and (3) transferring the material to another bin when the *average* moisture content for the mass is that desired; the mass must be mixed uniformly when moved.

The performance can be estimated by the deep-bed procedure of sect. 11.8.

11.10. Continuous Gravity-Flow Driers. Granular materials that flow readily, permit air to flow through them, and are not damaged in handling, can be dried in a gravity-flow drier such as the column drier of Fig. 11.6. The wet material is placed in the hopper and flows by gravity between the perforated retaining walls and is discharged at the bottom by a continuously operating

metering valve. Heated air is forced across the column at right angles to the direction of grain motion. The column may be inclined to simplify construction or fitted with baffles to stir the mass as it progresses through the drier. Many designs are used, but the principle of operation in all cases is that of Fig. 11.6.

The continuous gravity-flow drier is moderately expensive particularly when the materials-handling equipment required must be charged to it. High buildings are usually required to house it because height is required to get capacity. It is, however, a most acceptable device where large capacities and long-season use are characteristics of the operation. Labor costs are low since handling is completely mechanical.

The capacity is directly proportional to the column width and material movement rate through the column. The retention time in the column is the drying time for the material as defined by its drying indices, the required moisture reduction, and state factors of the drying air. Since the retention time is fixed for each individual situation, drier capacity is proportional to height.

Column-drier performance may be estimated by the deep-bed procedure. Note that the example, sect. 11.8, is for conventional column-drier performance. If the column of grain is stirred as it moves, the thin-layer procedure may be used with acceptable accuracy.

11.11. Rotary Driers. Materials that are not free flowing and that are not damaged by continuous handling are usually dried in rotary driers. Chopped forage, fruit and vegetable residues to be dried for livestock feed, and fertilizer components are examples of materials dried in this manner.

The rotary drier has a high initial cost and requires more floor space per unit of capacity than either the batch or column drier. Consequently, it should not be used if the batch or column drier is suitable.

Agricultural rotary driers such as shown in Fig. 11.7 are direct fired and single-, double-, and triple-drum types. The multiple-drum types are preferred since the over-all length can be short; and heat losses by conduction and radiation small.

The inside of the drum may be fitted with flights that lift the material and shower it down through the heated air. Flight design varies with the material to be dried. Chains or other dividing devices may be fitted to the inside of the drum to divide

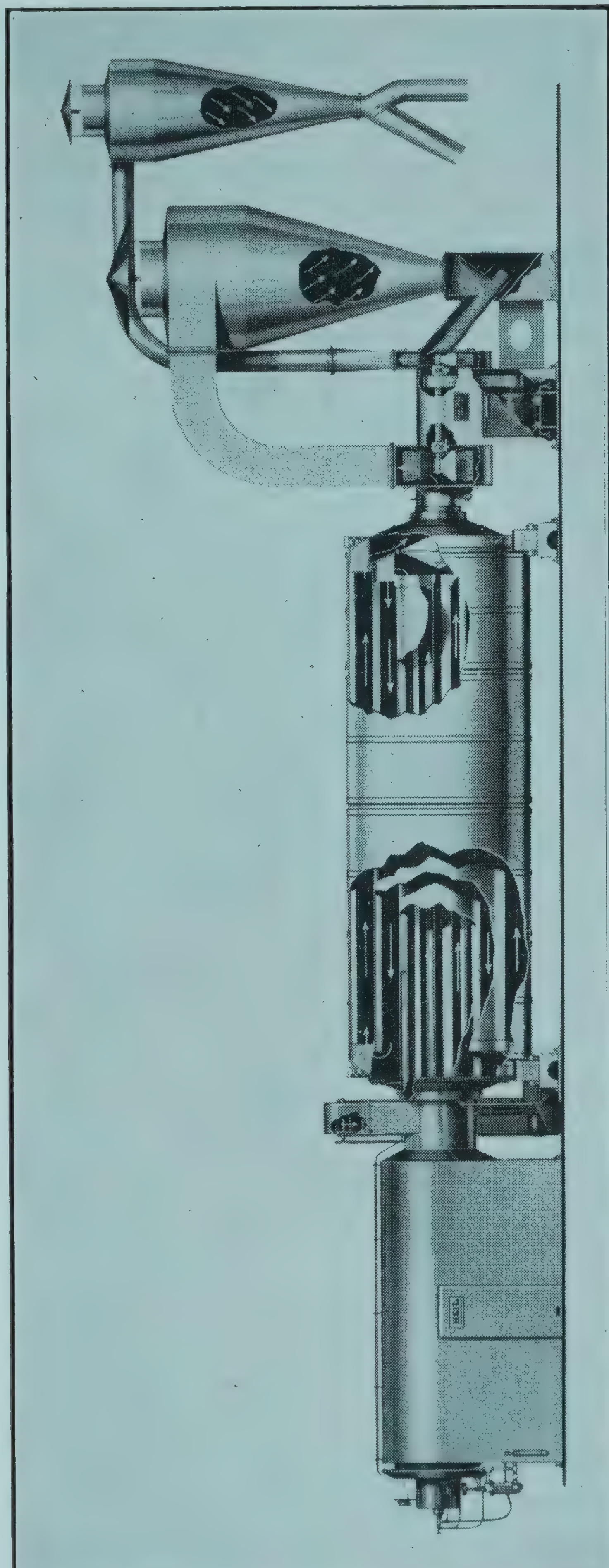


Fig. 11.7. A multiple-drum drier with a direct-fired heater and cyclones for collecting and cooling the dried product. (*Courtesy* The Heil Co.)

materials that tend to clump as they pass through the drier. The rate of material movement through the drum is controlled by flight design, inclining the drum, or by the rate of heated air through the drum, singly or combined. The drum should rotate at such a speed that the material is spilled uniformly through the cross-sectional space of the drum. This procedure yields a product of uniform final moisture content.

The capacity in pounds of material per hour depends upon the required reduction in moisture content, the drying indices for the material, the rate of air flow, and size of the drum.

Wet materials, e.g., green chopped alfalfa, particularly if covered with dew, and fruit and vegetable residues go through an initial constant-rate drying period where the material approaches (or reaches) the wet-bulb temperature. At the end of this period, falling-rate drying begins and the material temperature becomes progressively hotter than the wet-bulb temperature. Finely divided wet materials dry at a fast rate, and high air temperatures may be used. Temperatures as high as 1500°F may be used for chopped green alfalfa, for example.

Performance can be estimated by a stepwise thin-layer procedure. Consider a pound of dry material and its initial moisture. Dry through a delta time. Determine the moisture content and air state leaving the material at the end of the time element. The same material is dried through the second time element with air having a state that is the average of the initial and final leaving states for the first time element. This procedure is continued until the final moisture content is reached. The drum retention time and discharge air conditions are found. The air must discharge at a relative humidity below the equilibrium relative humidity of the material at its desired moisture content.

Control of the exhaust air state is frequently necessary. Since the air states can be defined by H, the following balance holds:

$$\frac{60\,VA}{v}(H_e - H_i) = \frac{W_d}{\theta} \cdot \frac{M_i - M_f}{100} \qquad (11.27)$$

where H_e = exhaust air humidity.
H_i = initial air humidity.
M_i = initial moisture content.
M_f = final moisture content.

Note that this balance is applicable for both counter- and parallel-flow drying. The stepwise calculation procedure requires that the delta humidity be added for parallel flow and subtracted for counterflow.

11.12. Tray Driers. Materials that cannot be dried by any of the previously discussed methods are dried on trays. Fruits

Fig. 11.8. Car of trays being moved into a tunnel drier. Note the tracks used to guide the cars. (*Courtesy* California Prune and Apricot Growers Association.)

and vegetables are best examples. The material is placed in shallow trays which are stacked on cars as shown in Fig. 11.8. The trays are spaced to permit air to circulate between them. The car of trays is dried in a cabinet or in a tunnel. Cabinet drying is a batch process with the principles applicable for calculations and is used for low-rate installations. Larger capacities are provided by tunnel systems, Fig. 11.9. The cars are moved through the tunnel by a slowly moving drag chain, a ratchet ram, or manually.

Parallel air flow gives a fast initial drying rate. Counterflow gives faster drying at the dry end of the tunnel. Parallel flow is seldom used because of its poor drying ability at the dry end of the tunnel. Combination tunnels utilize the advantages of

both parallel flow and counterflow, but the initial cost is greater and control is more difficult. Counterflow tunnels are most extensively used. The air rate must be high enough so that the relative humidity of the discharge air is below the equilibrium relative humidity of the material at the point where the material discharges.

Performance can be estimated by the procedure of rotary driers; a car and its contents is used as the unit of calculation. Note that the humidity change of the air due to the removal of moisture from the material is positive for parallel flow and negative

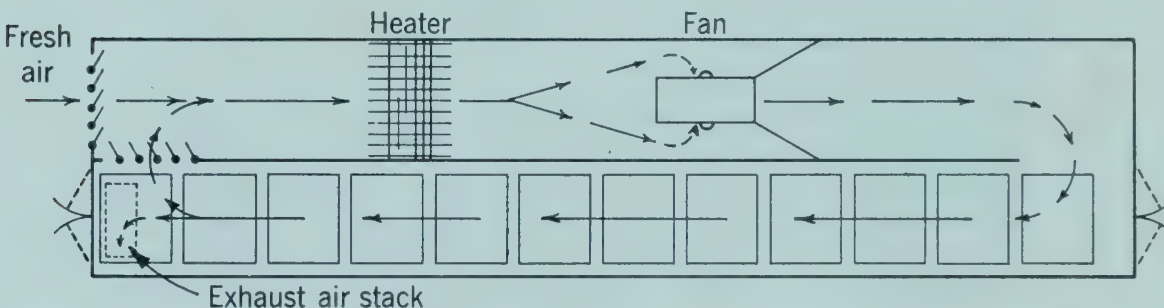


Fig. 11.9. Plan of a tunnel drier. The cars move from left to right for counterflow operation, from right to left for parallel flow.

for counterflow. The material-charging end is used as a reference in both cases.

11.13. Spray Driers. Spray driers remove the water from solutions or suspensions and dry the resulting powder to a moisture content that approaches equilibrium with the exhaust drying air. Spray driers are used extensively in the food, chemical, and pharmaceutical industries.

Design varies from a rectangular chamber fitted with spray jets, through which the drying air passes to continuous large-volume systems such as Fig. 11.10. Three procedures are used for breaking the material into fine drops.

1. *High-Pressure Atomization.* The liquid is forced through a nozzle under high pressure. Mixing with the drying air and the spray pattern can be controlled. Drop size and gradation are difficult to predict. Nozzle life is short when abrasive materials are sprayed.

2. *Centrifugal.* The liquid is fed at low pressure onto a horizontal disc or cup turning at speeds up to 20,000 rpm or more. The material breaks up into small drops as it leaves the edge of

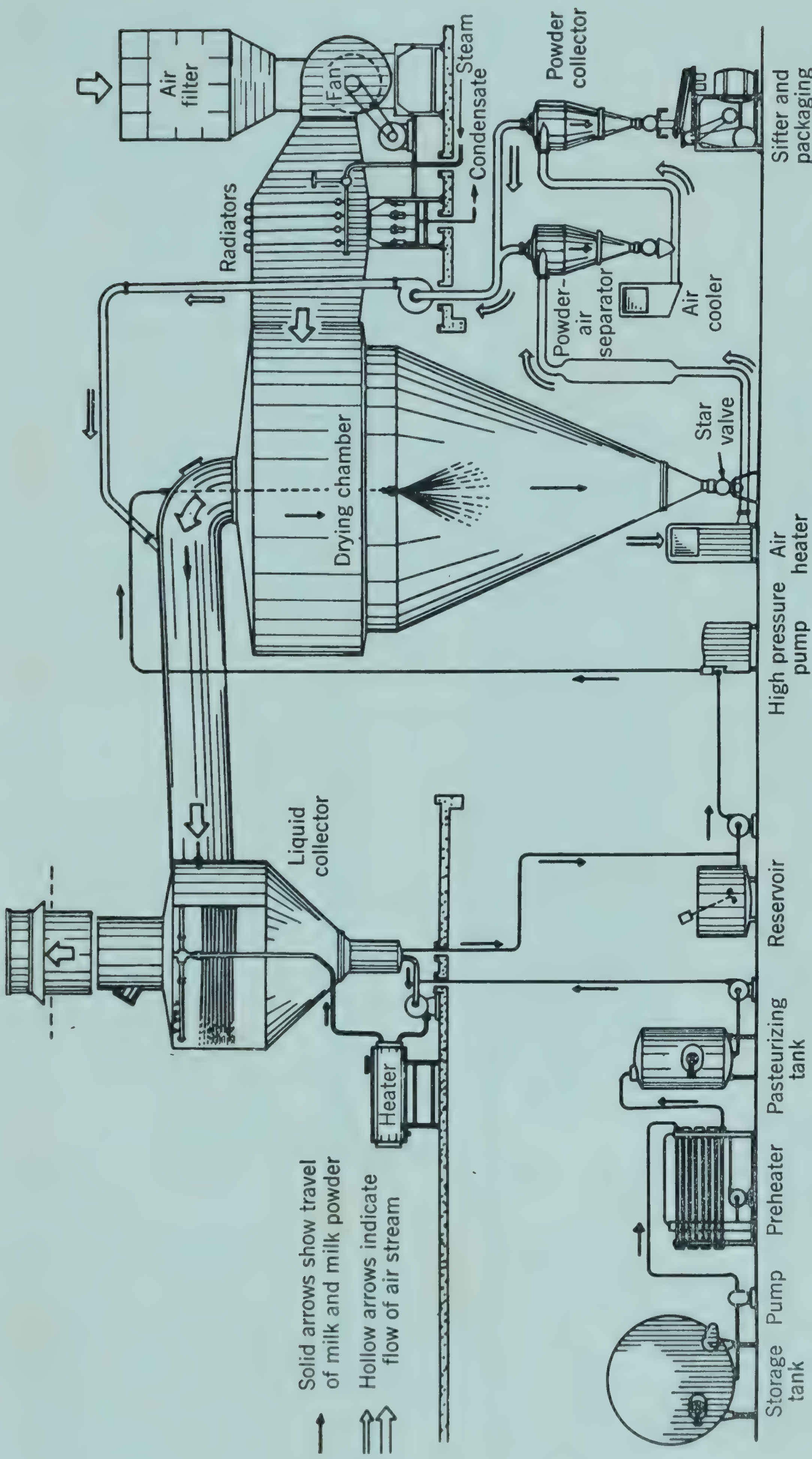


Fig. 11.10. A continuous-flow spray drier for milk products. (*Courtesy* Swenson Evaporator Co.)

the rotor. The drops are of uniform size, and materials not suitable for nozzles can be dried. Air-liquid drop mixing may be

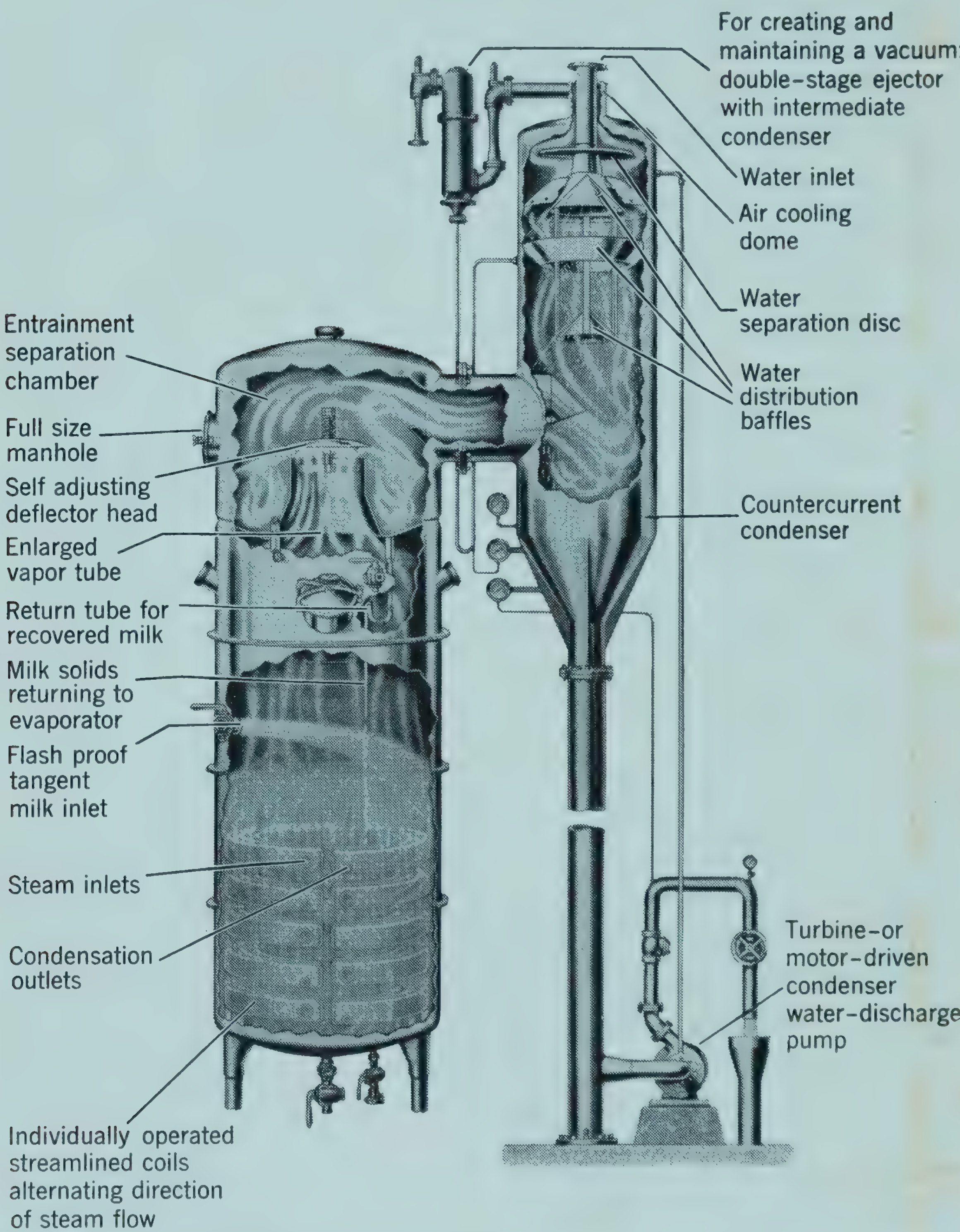
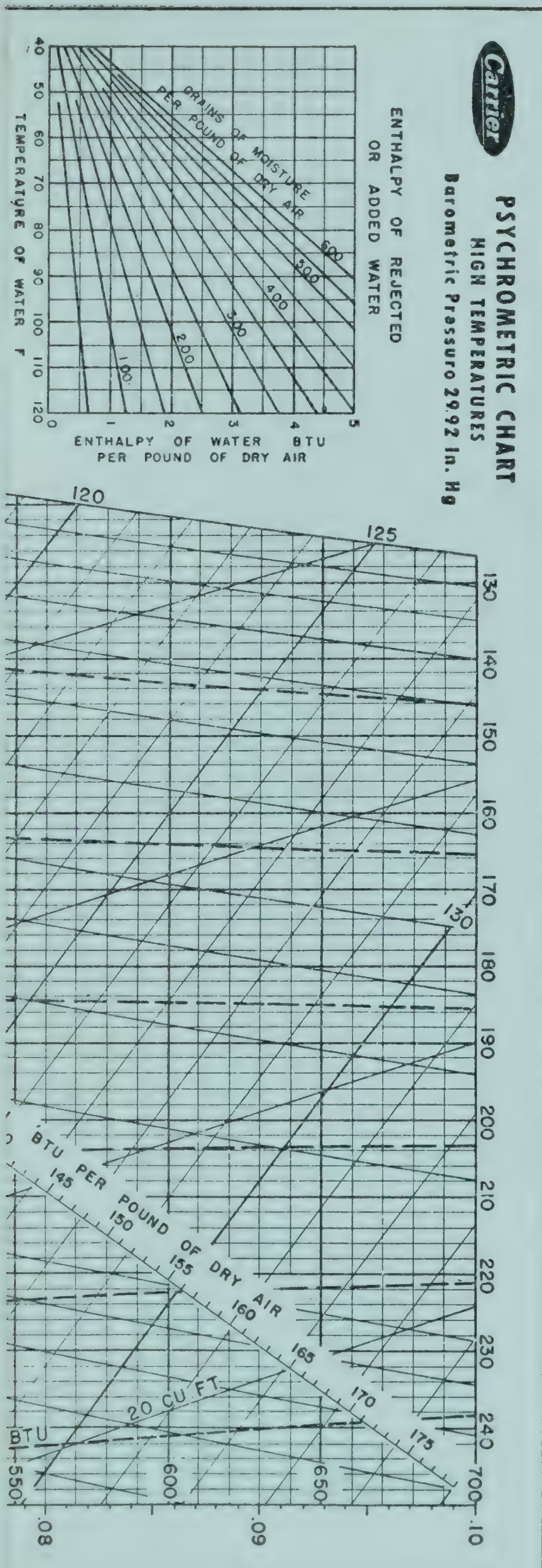


Fig. 11.11. Vacuum pan with entrainment separation chamber and countercurrent condenser used for milk concentration. (*Courtesy* Arthur Harris and Co.)

poor since the drops follow an umbrella-shaped trajectory. Disc-rotor and bearing maintenance are high.

3. *Two-Fluid Atomizing.* Air or steam under pressure breaks the liquid into fine drops by a mechanism comparable to that of


Carrier
PSYCHROMETRIC CHART
HIGH TEMPERATURES
Barometric Pressure 29.92 In. Hg
ENTHALPY OF REJECTED
OR ADDED WATER
GRAINS OF MOISTURE
PER POUND OF DRY AIR
TEMPERATURE OF WATER F
ENTHALPY OF WATER BTU
PER POUND OF DRY AIR
BTU PER POUND OF DRY AIR
20 CU FT
BTU

Corp.)

paint sprayers. Operating costs are high. This system is used only for the most difficult atomizing jobs and experimental units.

Because of the small diameter of the drops, drying is extremely fast and the material is dry when it reaches the walls or bottom of the container.

11.14. Concentrators. Concentrators, also called evaporators, are used to concentrate milk, fruit and vegetable juices, jams and jellies, etc., by boiling off a portion of the water. Because of the conditions of operation, the concentrator is frequently called a vacuum pan. A "pan" shown in Fig. 11.11 * is operated under a partial vacuum because (1) low-boiling temperatures do not damage heat-sensitive materials and (2) a large temperature difference is maintained between the steam and boiling liquid which permits a high heat-transfer rate.

The water spray condenses the water vapor removed from the concentrating liquid. The boiling temperature of the liquid being concentrated is controlled by the temperature of the water-condensate mixture leaving the condenser. The water-condensate mixture is removed by a pump or barometric leg. The vacuum is usually maintained by a steam ejector, although a vacuum pump may be used. Since the condenser handles the vapor from the concentrating liquid, the ejector needs to handle only the non-condensable gases and air from leaking gaskets.

The steam coils are usually fed with steam at 5 to 10 lb per sq in. The boiling is extremely vigorous because of the high temperature difference. Thus, a high heat rate is possible with minimum heat exchange surface and a "cooked" flavor is improbable owing to the surface speeds of the liquid.

This unit can operate on a batch basis or continuously by means of a suitable pump that continuously removes liquid from the bottom of the "pan" and by continuous feeding in of the liquid stock.

REFERENCES

1. Comings, E. W., and Sherwood, T. K. The Drying of Solids, VII. Moisture Movement by Capillarity in Drying Granular Solids. *Ind. and Eng. Chem.* 26:1096–98. 1934.

* This is only one of a number of designs which vary from company to company and for material to material.

2. Gamson, B. W., Thodos, G., and Hougen, O. A. Heat, Mass, and Momentum Transfer in the Flow of Gases Through Granular Solids. *Trans. Am. Inst. Chem. Engrs.* 39:1–35. 1943.
3. Gilliland, E. R., and Sherwood, T. K. The Drying of Solids, VI. Diffusion Equations for the Period of Constant Drying Rate. *Ind. and Eng. Chem.* 25:1134–36. 1933.
4. Henderson, S. M. A Basic Concept of Equilibrium Moisture. *Agr. Eng.* 33:29–32. 1952.
5. Hlynka, F., et al. A Comparative Study of Ten Electrical Meters for Determining Moisture Content of Wheat. *Canadian J. Research* 27:382–397. 1949.
6. Hougen, O. A., McCauley, H. J., and Marshall, W. R., Jr. Limitations of Diffusion Equations in Drying. *Trans. Am. Inst. Chem. Engr.* 36:183–209. 1940.
7. Marshall, W. R., Jr., and Hougen, O. A. Drying of Solids by Through Circulation. *Trans. Am. Inst. Chem. Engr.* 38:91–121. 1942.
8. Newman, A. B. The Drying of Porous Solids. Diffusion and Surface Emission Equations. *Trans. Am. Inst. Chem. Engr.* 27:203–310. 1931.
9. *Official and Tentative Methods of Analysis of the Association of Official Agricultural Chemists.* Published by the Association of Official Agricultural Chemists, P. O. Box 540, Benjamin Franklin Station, Washington, D. C.
10. Perry, R. L., et al., Fruit Dehydration. I. Principles and Equipment. *California Agr. Exp. Sta. Bull.* 698. 1946.
11. Perry, R. L. Heat and Vapor Transfer in the Dehydration of Prunes. *Trans. A.S.M.E.* 66:447–456. 1944.
12. Seltzer, E., and Settelmeyer, J. T. *Spray Drying of Foods. Advances in Food Research,* Vol. 2. Pp. 399–520. Academic Press. 1949.
13. Shedd, C. K. Mechanical Drying of Corn on the Farm. *U.S.D.A. Circ.* 839. 1950.
14. Sherwood, T. K. The Air Drying of Solids. *Trans. Am. Inst. Chem. Engr.* 32:150–68. 1936.
15. Sherwood, T. K. The Drying of Solids, II. *Ind. and Eng. Chem.* 21:976–980. 1929.
16. Sherwood, T. K. The Drying of Solids, III. Mechanism of the Drying of Pulp and Paper. *Ind. and Eng. Chem.* 22:132–36. 1930.
17. Sherwood, T. K. The Drying of Solids, IV. Application of Diffusion Equations. *Ind. and Eng. Chem.* 24:307–310. 1932.
18. Sherwood, T. K., and Comings, E. W. The Drying of Solids, V. Mechanism of Drying of Clays. *Ind. and Eng. Chem.* 25:311–16. 1933.
19. Van Arsdel, W. B. Approximate Diffusion Calculations for the Falling-Rate Phase of Drying. *Chem. Eng. Progr.* 43:13–24. 1947.
20. Von Loesecke, H. W. *Drying and Dehydration of Foods.* Reinhold. 1943.
21. Weisselberg, A., and Olive, T. R. Principles and Practices in Drying Solids. *Chem. Met. Eng.* 49:93–102. 1942.

PROBLEMS

1. 60,000 lb of shelled corn at 22 per cent moisture content (wet basis) are to be dried to 13 per cent moisture content. Determine:

   a. The dry-basis moisture contents.
   b. The amount of dry material.
   c. The gallons of water to be removed.

2. A thin layer of material dried with air of 110°F dry bulb and 80°F wet bulb had moisture contents at 2-hr intervals (dry basis) as follows: 30.0, 22.0, 17.3, 14.6, 13.1, and 12.2.

   a. Determine the drying constant α' and the apparent equilibrium moisture content of approach.
   b. How much time would be required to dry from 20 to 12 per cent (dry basis) with 110°F air? with 140°F air? Assume an equilibrium moisture content of 10 per cent (dry basis) and the same air rate as in (*a*).

3. A steel bin 14 ft in diameter and 8 ft deep contains grain sorghum which must be dried from 18 to 14 per cent (wet basis) moisture content. The floor is perforated, and air moves vertically through the mass at 2 cu ft per min per bu. Unheated 75°F air with a 55°F dew point is used for drying. If the average discharge air temperature is 66°F, how many hours will be required to dry the grain?
4. Develop a drying pattern similar to that of sect. 11.8 for an air rate of 100 cu ft per min and initial temperature of 160°F, the initial humidity being the same as that of the example.
5. Eggs are to be spray dried to 5 per cent (wet basis) moisture content. Air initially at 70°F dry-bulb temperature and 60°F wet-bulb temperature is heated to 250°F.

   a. What is the lowest possible discharge air temperature?
   b. How many British thermal units of heat must be supplied to the air per pound of moisture removed if the air discharges at 195°F?

6. A rotary drier uses 70°F, 36 per cent relative humidity air. It is heated to 250°F and discharges at 120°F. What per cent of the fuel might be saved if the latent and sensible heat in the discharge air were used to assist in heating the incoming air?
7. A tunnel dehydrator (Fig. 11.9) operates with a wet bulb controlled at 115°F. Outside air at 70°F and 60°F wet bulb is heated to 160°F. The air is discharged at 140°F. What per cent of the air must be recirculated?
8. Derive equation 11.3.

CHAPTER 12

Refrigeration

NOMENCLATURE

A = evaporator heat-exchanger area, sq ft.
c = specific heat, Btu per (°F lb).
D = piston displacement, cu ft per revolution.
E_c = compression thermal efficiency, per cent.
E_v = compressor volumetric efficiency, per cent.
h_a = heat content, vapor entering compressor, Btu per lb.
h_b = heat content, vapor leaving compressor, Btu per lb.
h_e = heat content, liquid entering evaporator, Btu per lb.
$h_f, h_{f'}$ = heat content, wet evaporating liquid, Btu per lb.
h_g = heat content, saturated vapor at evaporating pressure, Btu per lb.
$h_{g'}$ = heat content, superheated vapor at evaporating pressure, Btu per lb.
N = number of pistons.
q = heat rate, Btu per hr.
T_C = absolute temperature of evaporating liquid, °R.
T_H = absolute temperature of condensing liquid, °R.
t_1 = temperature of medium before cooling, °F.
t_2 = temperature of medium after cooling, °F.
t_3 = evaporating refrigerant temperature, °F.
U = heat-transfer coefficient, Btu per (hr sq ft °F).
v_g = specific volume of refrigerant vapor, cu ft per lb.
W_1, W_2, W_r = refrigerant rate, lb per hr.
W_m = mass rate, cooling medium, lb per hr.

Refrigeration may be defined as the process of removing heat from a body that is below the temperature of its surroundings. Or refrigeration may be defined as the process of transferring heat energy from a lower to a higher temperature. Natural refrigeration is that produced by the use of natural ice. Mechanical refrigeration is that accomplished by means of refrigerating engines which operate on thermodynamic principles.

12.1. Natural Refrigeration. Ice is satisfactory as a refrigeration medium for temperatures down to approximately 40°F

under such conditions as (1) short annual refrigeration periods, (2) duty away from power sources, and (3) where the ice cost is nominal. Temperatures below 32°F may be produced by mixing finely divided ice and various chemicals some of which are given in Table 12.1.

Table 12.1 TEMPERATURE OF ICE-FREEZING MIXTURES

Chemical	*Per Cent of Chemical in Mixture, by Weight*	*Temperature, °F*
NaCl	25	−1.6
$CaCl_2$	60	−27.6
HNO_3 (dilute)	50	−31.0
KOH	57	−38.3
HNO_3 (trace H_2SO_4)	50	−40.0

The latent heat of ice is 144 Btu per lb, and its specific heat is 0.47 Btu per lb per °F. The cooling rate is dependent upon the temperature difference and method of air or water circulation over it. Where ice is used for cooling recirculating water, it is difficult to secure water temperatures below 39°F if the ice is floating in a tank, because the maximum density of water is reached at 39°F and circulation within the tank is poor. A shower of return water over unsubmerged blocks of ice on a rack is much more effective than a tank.

12.2. Mechanical Refrigeration. Refrigeration processes using mechanical devices and electrical or other energy are called mechanical refrigeration systems. Two broad classifications are (1) absorption and (2) vapor compression systems. The vapor compression systems are the more common and will be discussed in this chapter.

The operation of the vapor compression system is shown schematically in Fig. 12.1. The liquid refrigerant in the receiver or supply tank is under high pressure. Because of this pressure the liquid is forced through the liquid line to and through the expansion valve into a region of low pressure produced by the compressor. The liquid refrigerant evaporates or boils to a vapor in the evaporator. The heat required for evaporation comes from the surroundings, and cooling results. The vapor moves at low

pressure through the vapor line to the compressor, is compressed to a high pressure, and passes to the condenser. Here it returns to the liquid state as the latent heat is transferred to the surroundings. The liquid then flows into the receiver.

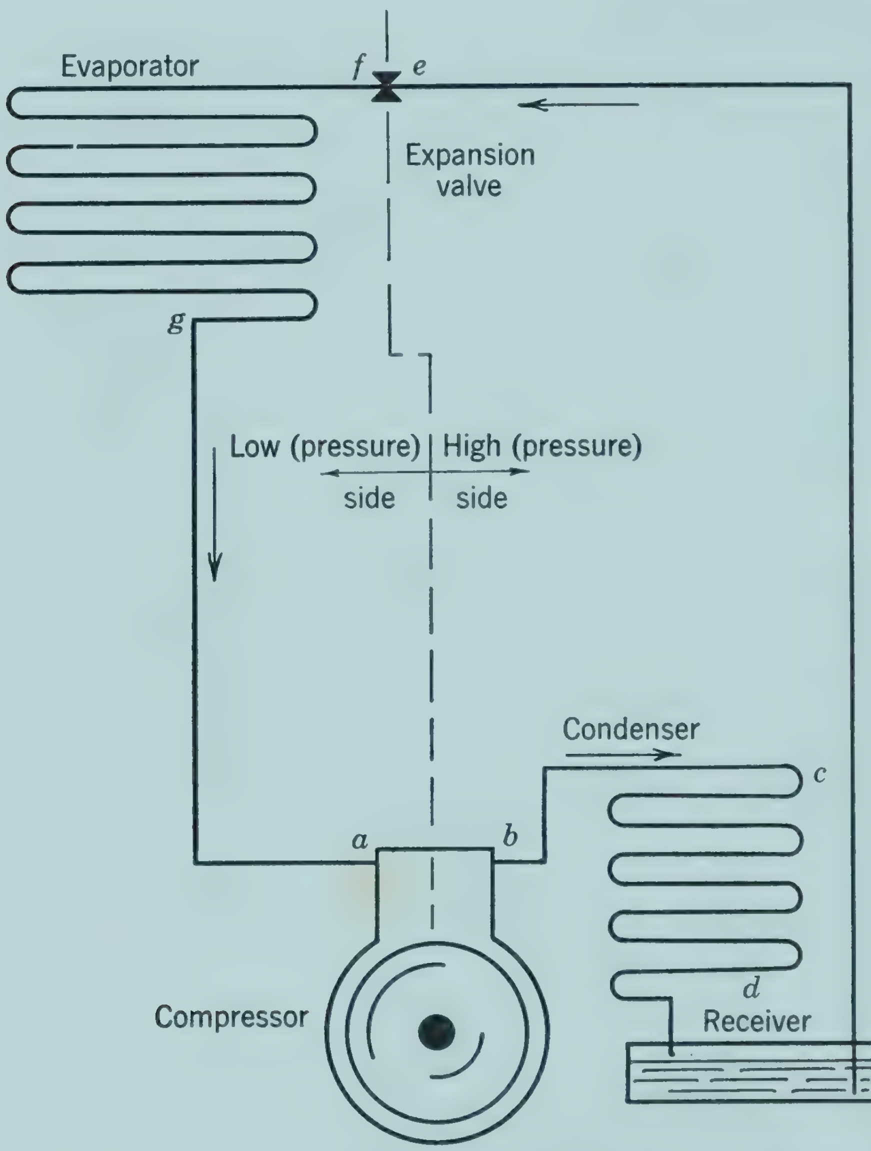


Fig. 12.1. A vapor compression refrigeration system.

The high-pressure side, called high side in the trade, is that to the right of the dotted line. The low-pressure side or low side is to the left of the dotted line.

Thermodynamically, the process can be shown explicitly by a Mollier (pressure-enthalpy) chart, Fig. 12.2, and the schematic system of Fig. 12.1. Liquid refrigerant in the receiver at a state between d and e flows toward the expansion valve and is subcooled to e by the surrounding air. An irreversible adiabatic process results when the liquid passes through the expansion valve.

The state changes from e to f, a part of the liquid flashing to a gas. The portion flashing is

$$(h_f - h_{f'})/(h_g - h_{f'}) \tag{12.1}$$

The wet mixture f evaporates to a state g' between g and a in the evaporator where the useful cooling takes place. The vapor further superheats to state a as it is conducted to the compressor.

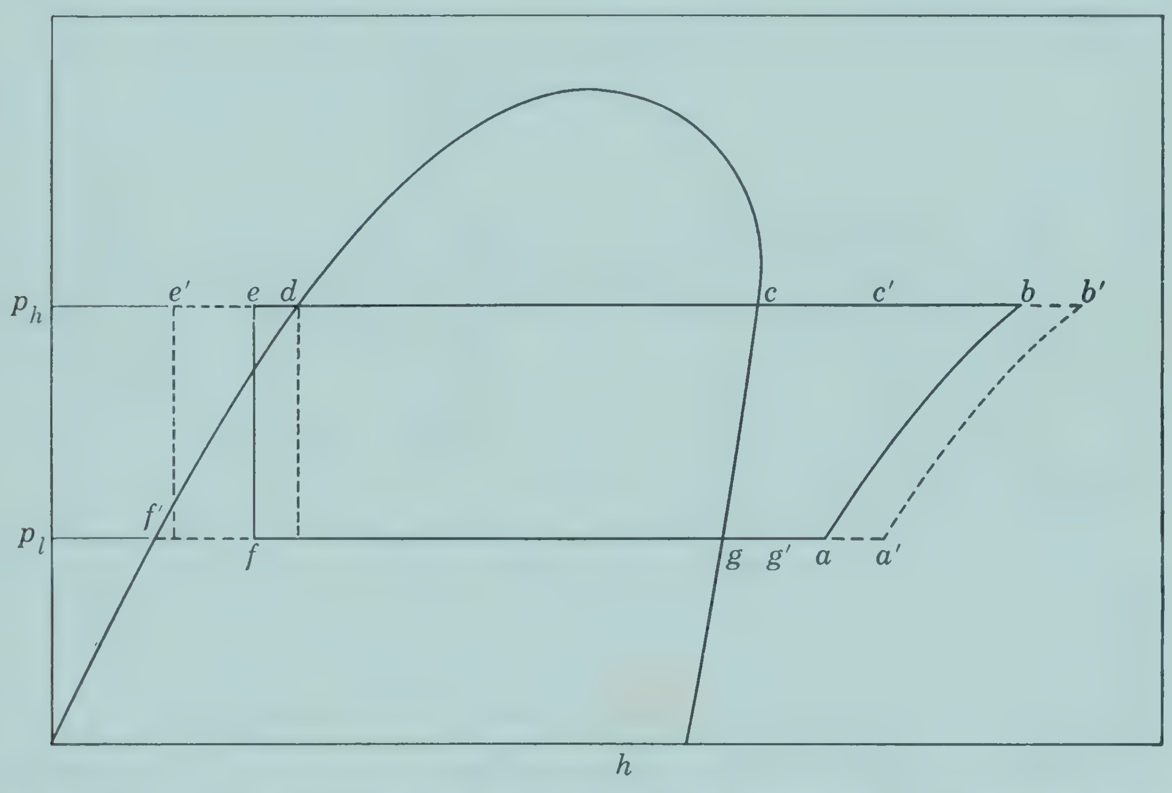


Fig. 12.2. The vapor compression mechanical refrigeration process, given schematically, by a Mollier (pressure-enthalpy) chart.

The gas is compressed isentropically by the compressor from a to b. The hot compressed gas is desuperheated to c and condensed to a liquid by removing the latent heat by the condenser, thus reaching point d. The liquid is usually subcooled to a point e in the condenser or by ambient air exchange.

The useful cooling or refrigeration per pound of refrigerant is

$$h_{g'} - h_f \tag{12.2}$$

The power required by the compressor per pound of refrigerant is $h_b - h_a$, which is the mechanical equivalent in terms of Btu. The coefficient of performance is a factor that designates the number of useful Btu of cooling capacity per equivalent mechanical Btu

input. The refrigeration cycle coefficient of performance, abbreviated as c.o.p., is:

$$\text{c.o.p.}_r = \frac{h_{g'} - h_f}{h_b - h_a} \tag{12.3}$$

The coefficient of performance relates the useful refrigerating energy to the mechanical energy input.

The Carnot or theoretical coefficient of performance for refrigeration is

$$\text{c.o.p.}_t = \frac{T_C}{T_H - T_C} \tag{12.4}$$

The temperatures are absolute; C refers to the temperature of the cold evaporating refrigerant; H, to the temperature of the hot condensing refrigerant. The actual c.o.p. is always smaller than the Carnot c.o.p. This results from the mechanical and thermal losses of a mechanical system and the characteristics of the refrigeration cycle.

The coefficient of performance will range from less than one for systems operating at subzero temperatures to five or more for systems operating above freezing; the smaller the difference between p_h and p, the greater the coefficient of performance.

12.3. Rating. Refrigerating systems and components are rated on the basis of tons or Btu per hr. The term "ton" originated when mechanical refrigeration was in its infancy and was used for comparing the performance of mechanical systems with ice. A ton of ice absorbs 144 × 2000 or 288,000 Btu in melting (in providing refrigeration). A machine that can absorb heat (produce refrigeration) at the rate of 288,000 Btu per day is rated at 1 ton. In making ice, water must be cooled from some ambient temperature, say 70°F, and the ice is in practice finished considerably below 32°F, thus a 1-ton machine can make only about ⅔ ton in 24 hr. One ton is equal to 12,000 Btu per hr, or 200 Btu per min. Common practice is to rate small systems on a Btu-per-hr basis, large systems in tons. The evaporator temperature should be specified since the capacity decreases as the temperature of the evaporating refrigerant decreases.

The American Society of Refrigerating Engineers has adopted certain "standard operating conditions" which facilitate the com-

Table 12.2 SOME PROPERTIES OF REFRIGERANTS ARRANGED IN ORDER OF THE BOILING POINT AT ATMOSPHERIC PRESSURE

Refrigerant	*Formula*	*Boiling Point, °F*	*Saturated Pressure, psig*		*Refrigeration per lb, $h_g5° - h_f86°F$*	*Vapor Rate, cfm per ton at 5°F*	*Toxicity No.**	*Inflammable Limits, by Volume in Air, %*
			5°F	*86°F*				
Water	H_2O	212	29.9 †	28.6 †	1007.3	23,000	6	Nonflammable
Freon-113	$CCl_2F\text{-}CClF_2$	117.6	27.9 †	13.9 †	53.7	101.0	4	Nonflammable
Methylene chloride	CH_2Cl_2	104.9	27.5 †	9.4 †	134.6	74.0	4	Nonflammable
Freon-11	CCl_3F	74.7	24.0 †	3.6	67.5	36.3	5	Nonflammable
Freon-21	$CHCl_2F$	48.0	19.3 †	16.5	89.4	20.4	–	Nonflammable
Freon-114	$CClF_2$	38.4	16.1 †	22.0	43.1	19.6	6	Nonflammable
Butane	C_4H_{10}	30.9	13.2 †	26.7	123.7	16.1	5	1.6–6.5
Sulfur dioxide	SO_2	13.6	5.9 †	51.8	141.4	9.1	1	Nonflammable
Methyl chloride	CH_3Cl	−10.8	6.4	80.0	150.3	5.9	4	8.1–17.2
Freon-12	CCl_2F_2	−21.6	11.8	93.2	51.1	5.8	6	Nonflammable
Ammonia	NH_3	−28.0	19.6	154.5	474.4	3.4	2	16–25
Freon-22	$CHClF_2$	−41.4	28.3	159.8	69.3	3.6	5	Nonflammable
Propane	C_3H_8	−44.2	27.2	140.6	121.0	4.1	5	2.3–7.3
Carbon dioxide	CO_2	−109.2 ‡	322	1031	55.5	1.0	5	Nonflammable
Nitrous oxide	N_2O	−127.0	294	922	50.3	1.1	–	Nonflammable

* From the National Board of Fire Underwriters; No. 1 is most toxic, No. 6 harmless.
† Vacuum, in. mercury.
‡ Sublimation temperature of solid.

parison of refrigerants, systems, and components. These conditions are:

Refrigerant evaporation temperature, 5°F.
Refrigerant condensing temperature, 86°F.
Superheating, evaporator to compressor, 9°F.
Supercooling, condenser to expansion valve, 9°F.

A number of refrigerants compared on this basis are listed in Table 12.2.

The boiling point is a general indication of the temperature at which the refrigerant would be used. The lower the boiling point (saturated temperature at 0 psi gage), the lower the service temperature. For example, F-22 is preferred over F-12 for temperatures below −22°F.

The gage pressures at 5°F and 86°F further assist in determining the temperature operating level since a high pressure at 5°F implies that a reduction in pressure will effect a lower evaporating temperature. The pressure at 86°F is an indication of the type of design required such as line joints, shaft seals, compressors.

The refrigeration per pound of refrigerant is an inverse index of the required rate of liquid flow and the size of liquid lines needed.

The vapor rate per ton determines the volumetric capacity of the compressor and the size of the vapor lines.

12.4. General Considerations. Other system characteristics that are important in selecting a refrigerant for a job, changing refrigerants in an installation, or selecting the equipment are:

1. *Chemical Reactions.* Sulfur dioxide will not attack steel or copper if dry. If moisture is present sulfurous acid may be formed and both steel and copper and related materials will be attacked. Ammonia will not attack iron or steel even if water is present. Copper and related alloys are not attacked by dry ammonia. However, their use is not recommended since a perfectly anhydrous ammonia refrigerant is most difficult to maintain. The other refrigerants in Table 12.2 are essentially chemically inert.

2. *Moisture in System.* Water is sufficiently soluble in ammonia, carbon dioxide, and sulfur dioxide that moderate amounts can move within the system without freezing occurring in the low

Table 12.3 SOME PROPERTIES OF SATURATED AMMONIA

Temperature, °F	*Pressure, psi*		*Vapor Volume, cu ft/lb*	*Enthalpy, Btu/lb*	
	Absolute	*Gage*		*Liquid*	*Vapor*
−50	7.67	14.3 *	33.08	−10.6	593.7
−40	10.41	8.7 *	24.86	0.0	597.6
−30	13.90	1.6 *	18.97	10.7	601.4
−28	14.71	0.00	18.00	12.8	602.1
−26	15.55	0.8	17.09	14.9	602.8
−24	16.42	1.7	16.24	17.1	603.6
−22	17.34	2.6	15.43	19.2	604.3
−20	18.30	3.6	14.68	21.4	605.0
−18	19.30	4.6	13.97	23.5	605.7
−16	20.34	5.6	13.29	25.6	606.4
−14	21.43	6.7	12.66	27.8	607.1
−12	22.56	7.9	12.06	30.0	607.8
−10	23.74	9.0	11.50	32.1	608.5
−8	24.97	10.3	10.97	34.3	609.2
−6	26.26	11.6	10.47	36.4	609.8
−4	27.59	12.9	9.991	38.6	610.5
−2	28.98	14.3	9.541	40.7	611.1
0	30.42	15.7	9.116	42.9	611.8
2	31.92	17.2	8.714	45.1	612.4
4	33.47	18.8	8.333	47.2	613.0
5	34.27	19.6	8.150	48.3	613.3
6	35.09	20.4	7.971	49.4	613.6
8	36.77	22.1	7.629	51.6	614.3
10	38.51	23.8	7.304	53.8	614.9
12	40.31	25.6	6.996	56.0	615.5
14	42.18	27.5	6.703	58.2	616.1
16	44.12	29.4	6.425	60.3	616.6
18	46.13	31.4	6.161	62.5	617.2
20	48.21	33.5	5.910	64.7	617.8
22	50.36	35.7	5.671	66.9	618.3
24	52.59	37.9	5.443	69.1	618.9
26	54.90	40.2	5.227	71.3	619.4
28	57.28	42.6	5.021	73.5	619.9
30	59.74	45.0	4.825	75.7	620.5
32	62.29	47.6	4.637	77.9	621.0
35	66.26	51.6	4.373	81.2	621.7
40	73.32	58.6	3.971	86.8	623.0
60	107.6	92.9	2.751	109.2	627.3
65	117.8	103.4	2.520	114.8	628.2
70	128.8	114.4	2.312	120.5	629.1
75	140.5	125.8	2.125	126.2	629.9
80	153.0	139.9	1.955	132.0	630.7
85	166.4	151.7	1.801	137.8	631.4
86	169.2	154.5	1.772	138.9	631.5
90	180.6	165.9	1.661	143.5	632.0
95	195.8	181.1	1.534	149.4	632.6
100	211.9	197.2	1.419	155.2	633.0
105	228.9	214.2	1.313	161.1	633.4
110	247.0	232.3	1.217	167.0	633.7
115	266.2	251.5	1.128	173.0	633.9
120	286.4	271.7	1.047	179.0	634.0

* In. mercury below 1 atmosphere.

Table 12.4 SOME PROPERTIES OF SATURATED FREON-12

Temperature, °F	*Pressure, psi*		*Vapor Volume, cu ft/lb*	*Enthalpy, Btu/lb*	
	Absolute	*Gage*		*Liquid*	*Vapor*
−50	7.125	15.42 *	5.012	−2.11	72.31
−40	9.317	10.96 *	3.911	0.00	73.50
−30	12.02	5.45 *	3.088	2.03	74.70
−28	12.62	4.23 *	2.950	2.44	74.94
−26	13.26	2.93 *	2.820	2.85	75.18
−24	13.90	1.63 *	2.698	3.25	75.41
−22	14.58	0.24 *	2.583	3.66	75.64
−20	15.28	0.58	2.474	4.07	75.87
−18	16.01	1.31	2.370	4.48	76.11
−16	16.77	2.07	2.271	4.89	76.34
−14	17.55	2.85	2.177	5.30	76.57
−12	18.37	3.67	2.088	5.72	76.81
−10	19.20	4.50	2.003	6.14	77.05
−8	20.08	5.38	1.922	6.57	77.29
−6	20.98	6.28	1.845	6.99	77.52
−4	21.91	7.21	1.772	7.41	77.75
−2	22.87	8.17	1.703	7.83	77.98
0	23.87	9.17	1.637	8.25	78.21
2	24.89	10.19	1.574	8.67	78.44
4	25.96	11.26	1.514	9.10	78.67
5	26.51	11.81	1.485	9.32	78.79
6	27.05	12.35	1.457	9.53	78.90
8	28.18	13.48	1.403	9.96	79.13
10	29.35	14.65	1.351	10.39	79.36
12	30.56	15.86	1.301	10.82	79.59
14	31.80	17.10	1.253	11.26	79.82
16	33.08	18.38	1.207	11.70	80.05
18	34.40	19.70	1.163	12.12	80.27
20	35.75	21.05	1.121	12.55	80.49
22	37.15	22.45	1.081	13.00	80.72
24	38.58	23.88	1.043	13.44	80.95
26	40.07	25.37	1.007	13.88	81.17
28	41.59	26.89	0.973	14.32	81.39
30	43.16	28.46	0.939	14.76	81.61
32	44.77	30.07	0.908	15.21	81.83
35	47.28	32.58	0.863	15.88	82.16
40	51.68	36.98	0.792	17.00	82.71
60	72.41	57.71	0.575	21.57	84.82
65	78.43	63.74	0.532	22.72	85.32
70	84.82	70.12	0.493	23.90	85.82
75	91.60	76.90	0.458	25.08	86.32
80	98.76	84.06	0.425	26.28	86.80
85	106.4	91.7	0.395	27.48	87.28
86	107.9	93.2	0.389	27.72	87.37
90	114.3	99.6	0.368	28.70	87.74
95	122.8	108.0	0.343	29.93	88.19
100	131.6	116.9	0.319	31.16	88.62
105	140.9	126.2	0.293	32.40	89.03
110	150.7	136.0	0.277	33.65	89.43
115	161.0	146.3	0.258	34.90	89.80
120	171.8	157.1	0.240	36.16	90.15

* In. mercury below 1 atmosphere.

temperature regions. Water is essentially nonsoluble in the halide refrigerants, and even minute quantities may freeze in the expansion valve or capillary tube, shutting off the flow of refrigerant. A moisture-absorbing cartridge is usually inserted in the liquid refrigerant line to remove the moisture from the refrigerant or the system. Oil and refrigerant are thoroughly dried before assembling. Moisture accelerates the formation of sludge.

3. *Oil Miscibility.* Oil is not soluble with ammonia and carbon dioxide and has limited solubility in sulfur dioxide and nitrous oxide. Oil moves in these systems in drops or slugs and accumulates at low points in the system where it must be removed at periodic intervals. Oil is soluble in the halide and hydrocarbon refrigerants and moves as a solution. Circulation within the system is usually continuous, the oil moving through the vapor line as a fog. Difficulty may develop if oil-soluble refrigerants are used in flooded evaporator systems because of excessive concentration of oil by fractional evaporation of refrigerant.

Refrigerant tables and Mollier charts are comparable to steam tables and charts. Some data for ammonia and Freon-12 are given in Tables 12.3, 12.4, and 12.5 and Figs. 12.3, 12.4, and 12.5. For more complete data the *Refrigerating Data Book* or a refrigeration textbook should be consulted.

Table 12.5 SATURATED LIQUID REFRIGERANT DENSITIES

Temperature, °F	*Liquid Density, lb per cu ft*	
	Ammonia	*Freon-12*
−40	43.08	94.58
−20	42.22	92.58
0	41.34	90.52
20	40.43	88.37
40	39.49	86.10
60	38.50	83.78
80	37.48	81.39
100	36.40	78.80
120	35.26	76.02

COMPONENTS

12.5. Compressors. The compressor changes the gas from state a to state b, Figs. 12.1 and 12.2, and is characterized by the

volume rate of the gas at intake pressure and the pressure change affected. Four types of compressors are in general use.

1. *Reciprocating* or piston-type compressors (Fig. 12.8) are most extensively used. Small units directly connected to electric motors are used for household refrigerating systems. Piston diameters and strokes of less than an inch are common in the latter systems. Multicylinder units are used for large industrial systems. Reciprocating compressors are used for all refrigerants and exclusively for those operating at high-pressure differentials, for refrigerants in the lower portion of Table 12.2. The efficiencies (volumetric, thermal, and mechanical) are high.
2. *Rotary* compressors (Fig. 4.4) are used with some success in household and other small systems where pressure differentials are small or moderate. They are mechanically simpler than reciprocating units, are quiet, have high volumetric capacity with high rotative speeds, and consequently occupy small space. Starting torque is less than for reciprocating compressors since there is a smaller variation in pressure per rotative cycle. Tolerances must be exceptionally close to insure volumetric performance. Lubrication is a problem since the vanes or other gas-confining parts are usually spring or centrifugally loaded. Wear soon increases clearances, and the volumetric capacity decreases. The mechanical efficiency may be low because of internal friction.
3. *Gear* compressors (Fig. 4.1) have the same performance features as rotary compressors except that the volumetric efficiency may be less and the mechanical efficiency higher because of less starting friction. Their chief use is for boosting in compound systems.
4. *Centrifugal* compressors (Chap. 5) are used extensively for large systems using refrigerants with large specific vapor volume and small pressure differential. Air conditioning systems using refrigerants from the upper portion of Table 12.2 might employ centrifugal compressors. Single and multistaging is used, depending upon the pressure differential. Performance is comparable to a centrifugal air compressor. The volumetric capacity can be adjusted by throttling the discharge, a most convenient feature not possible with positive displacement units.

12.6. Condensers. The condenser cools the compressed gas to the saturation temperature and condenses it to a liquid, process

c'–d in Fig. 12.2. Some of the superheat, represented by b–c, may be removed by a special desuperheater located between the compressor and the condenser. This can be a water-cooled heat exchanger or simply a finned or extra-long bare pipe between the compressor and condenser which permits heat to escape to the room. The latent heat c–d is removed, and the liquid is subcooled to a point between e and d. Significant subcooling cannot take place as long as the vapor is in contact with the liquid. Subcooling, then, results only if the liquid is in a vapor-free heat-exchange region. Further subcooling, to e, takes place between the receiver and the evaporator.

Four types of condensers are common in agricultural work.

1. *Air-Cooled Condensers* which make use of finned tubes are used on systems up to 3 hp. Usual construction is vertical fins with horizontal tubes, the vapor being fed in at the top, the liquid flowing by gravity to the lower part of the condenser and thence to the receiver. Air is forced through the condenser by a fan.

2. *Shell and Tube Condensers* consist of a cylindrical drum with a series of water tubes inside. Large-capacity units are vertical, smaller units horizontal. The horizontal unit usually serves as a combination condenser and receiver. The water tubes are located in the upper portion of the cylinder so that the condensing surface will not be covered with liquid.

If the supply of water is ample and the cost low, the water is used but once and then discarded. If the supply is low or the cost high, the water may be circulated through a cooling tower where it is cooled by a portion evaporating. With a tower the actual water usage may be only 2 per cent of that where it is wasted.

3. *Combination Air- and Water-Cooled Condensers* are available for small systems that may be required to operate when air temperatures are high, 95°F or higher. The water flow is controlled by the high side-pressure so water is used only when high temperatures of the air cause high head-pressure.

4. *Evaporative Condensers* are extensively used where water supply (or disposal) or high temperatures are a problem. Water is recirculated over the pipes of the condenser in a thin film, spray, or shower. A forced draft of air over the wet pipes causes some of the water to evaporate. The heat liberated by the condensing

Forced circulation of the liquid by a pump may be employed to provide even better performance. Because of the vigorous action, drops of refrigerant are carried toward the vapor discharge port. These drops are separated from the spent vapor and collect in the "accumulator." The liquid-free gas then returns to the compressor. This system is not completely satisfactory for oil-soluble refrigerants. The gas discharge action is not vigorous enough to carry out the oil that accumulates in the unit in solution. Non-soluble refrigerants permit the oil to settle out, and it is drained off at periodic intervals.

(*e*) A *shell and tube evaporator* has the same operational characteristics as the accumulator system. It is used for cooling brine, water, or other liquids. The cooling material must not be permitted to freeze.

(*f*) The *ice-bank* evaporator is used where large quantities of "chilled" water at 32°F are needed. Evaporation takes place in a series of plates or bank of tubes which are immersed in a tank of water. During periods of low-water demand ice accumulates on the plates. The ice is then available for cooling the water during peak-demand periods. This procedure permits a smaller compressor to be used than would be required with the non-ice system.

(*g*) Evaporator performance can be improved by installing a *heat exchanger* or *regenerator* immediately following the evaporator. Thus, Fig. 12.2, state a is moved toward a' and e toward e', f in turn moving toward f'. This decreases the percentage of flash vapor at the expansion valve but increases the superheat at the compressor. Subcooling to e can be produced by locating the heat exchanger so its trailing end is in the liquid evaporating region.

12.8. Expansion Valves. The expansion valve is used to regulate the rate of flow of liquid refrigerant into the evaporator at the evaporating rate. Four types of valves are used.

1. *Manually Adjusted Needle Valves* may be used in large systems where loads are relatively constant and an attendant is on duty. Their advantageous features are quick adjustment, simplicity, and low first cost.

2. *Float Valves* are actually automatically adjusted needle valves since they are so positioned that the incoming liquid rate

equals the evaporating rate. They are used, as previously discussed, in flooded systems with accumulators.

3. *Capillary Tubes* are used extensively in household refrigeration and other small systems. They are suitable only for systems composed of a single compressor and a single evaporator. The liquid passes from the high to the low side through a small tube of such a diameter and length that the rate of flow at operating pressure does not exceed the evaporating capacity at the design load. The system is simple since there are no valves, except in the compressor, and a receiver is not used. The system elements

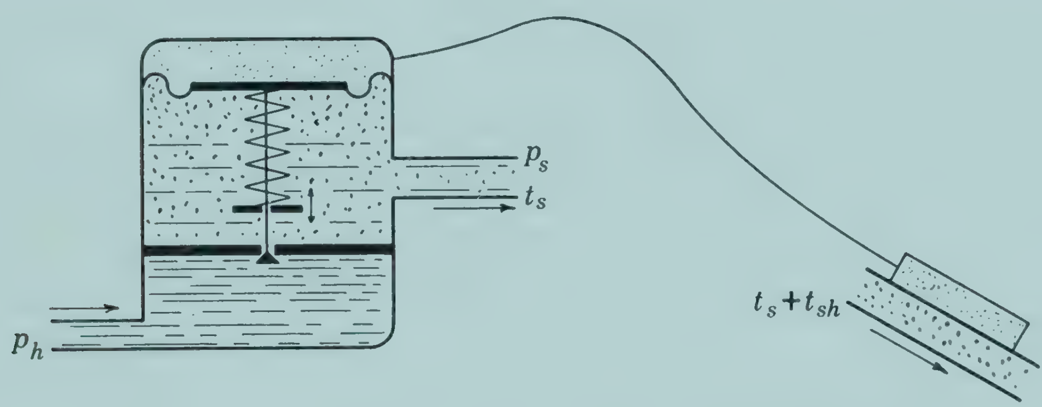


Fig. 12.7. Principles of the thermal expansion valve.

are located relative to each other so that when operation has ceased, the liquid will flow into the evaporator and/or condenser and pressures will equalize throughout the system. Consequently, there is no pressure differential across the compressor when it starts. This permits a low-starting torque motor to be used. The quantity of refrigerant charged into the system must be controlled carefully.

4. *Thermal Expansion Valve,* Fig. 12.7, operates on the basis of the number of degrees of superheat in the spent vapor leaving the evaporator. Thermal expansion valves are used on evaporators (a), (b), and (c), and on systems of a wide range of sizes and those with more than one evaporator. They are especially applicable for field-assembled systems with automatic control and variable-cooling load.

The thermal sensing bulb is usually filled with the same fluid used for the refrigerant. Consequently, the downward pressure on the diaphragm due to a temperature of the bulb is comparable to the upward pressure on the diaphragm due to the saturated

pressure of the refrigerant within the evaporator. This force referred to the spring reaction maintains a proper opening of the valve for all temperatures and loads, the design or adjusted superheat applying at all times. Consideration of these data and the physical characteristics of the valve will show that:

1. The superheat to the sensing bulb is constant at all loads. The temperature difference between the boiling liquid and the material being cooled may be greater than the superheat but never less.
2. The liquid rate is controlled on the basis of the heat load. Thus, the amount of evaporator surface used for heat exchange is controlled by the heat load.
3. Liquid flow is stopped when the compressor stops. This feature facilitates control which is discussed later in this chapter.

SYSTEM DESIGN AND BALANCE

The heat and mass balance for a system can be expressed by the following equation.

12.9. The Evaporator:

$$W_m c(t_1 - t_2) = AU(t_1 - t_3) = W_r(h_{g'} - h_f) = q \quad (12.5)$$

where W_m = mass rate of medium to be cooled, lb per hr.
c = specific heat of medium.
$t_1 - t_2$ = temperature drop of cooling medium.
W_r = refrigerant rate, lb per hr.
$h_{g'}$ = heat content of vapor leaving evaporator, Btu per lb.
h_f = heat content of the liquid refrigerant entering the evaporator, Btu per lb.
A = evaporator heat-exchange area, sq ft.
t_3 = temperature of the evaporating refrigerant, °F.
q = heat rate, Btu per hr.
U = heat-transfer coefficient, Btu per (hr sq ft °F between medium entering exchanger and refrigerant).

The refrigeration load is defined by (1) the heat rate q, (2) original and final temperature t_1 and t_2, and perhaps (3) relative humidity. Commercial evaporators are rated on a heat rate per degree temperature difference, T.D., $t_1 - t_3$ in equation 12.5.

The student should understand that a rigorous treatment of the heat-exchanger performance would require that the log-mean temperature difference be used. Such a difference would be used in analyzing a unit that was not test rated on the basis of t_1 and t_3. This rating procedure is used because of convenience. Cooling loads which can be carried with a large T.D. can be handled with an evaporator of small effective area A. Note however that a low evaporator temperature will require a low side-pressure and a compressor of greater volumetric capacity will be needed. The T.D. is usually limited by the possibility of freezing, frost formation, or dehumidification of the air.

12.10. Defrosting of evaporators for air cooling above 34°F is brought about by using an evaporator of sufficient size so that the load can be handled with only part-time operation. Frost accumulating during the running cycle is melted during the off-cycle. This procedure, in addition to defrosting, helps to maintain humidity if high humidities are desired.

Evaporators for below freezing temperatures are designed with fins a greater distance apart or with bare coils so that moderate frost or ice accumulation will not affect air flow. Defrosting is by hot water, electric heaters, or hot gas. Hot gas from the high side of the system is piped to the evaporator by a series of connecting lines and valves. Operation may be manual or automatic.

Frosting may be prevented by continuous circulation of a brine over the evaporating surface. The brine picks up moisture from the surface and must be reconcentrated or replaced at intervals.

12.11. The Compressor:

$$W_r v_g = 60 E_v D N \text{ rpm} \tag{12.6}$$

where v_g = specific volume of the vapor entering the compressor, at low side pressure, cu ft per lb.

E_v = compressor volumetric efficiency, 60 to 75 per cent for small reciprocating compressors.

D = piston displacement, cubic feet per piston revolution.

N = number of pistons.

rpm = compressor speed.

Horsepower per ton of refrigeration (thermal)

$$= \frac{200}{h_{g'} - h_e} \times \frac{(h_b - h_a)100}{E_c(33{,}000/778)} \tag{12.7}$$

where $h_{g'}$ = heat content of vapor leaving evaporator, Btu per lb.
h_e = heat content of liquid entering evaporator, Btu per lb.
h_b = heat content of vapor leaving compressor, Btu per lb.
h_a = heat content of vapor entering compressor, Btu per lb.
E_c = compression efficiency, 65 to 80 per cent (estimated).*

Equation 12.6 defines the capacity of a compressor in terms of its speed, its physical characteristics, and the mass rate of the

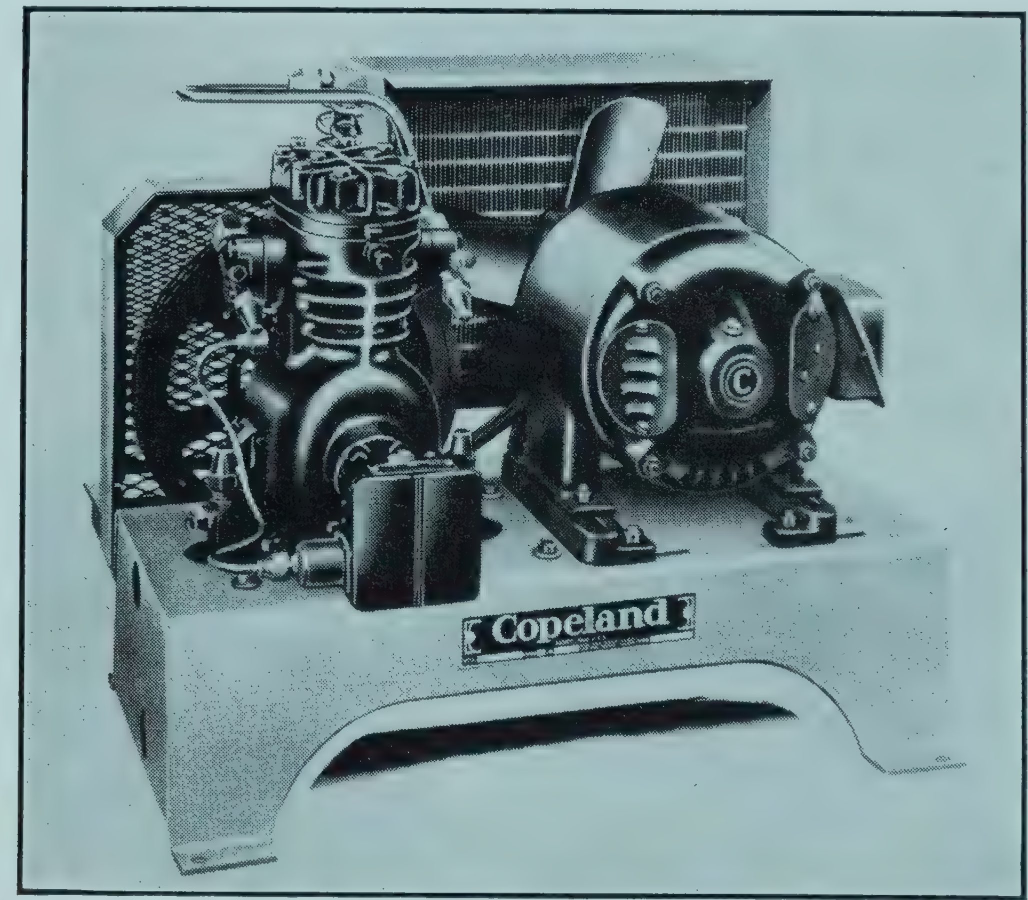


Fig. 12.8. Conventional compressor-condenser unit. (*Courtesy* the Copeland Co.)

refrigerant. The vapor volume rate represented by the product $W_r v_g$ must be the same as that for the conditions in equation 12.5 with corrections for line friction. The compressor capacity must be balanced with the capacity of the evaporator. A compressor with more capacity than that specified for the desired conditions will move a greater volume of vapor at a greater specific volume. This will lower the suction pressure and evaporating temperature and increase the mass evaporation rate. The

* The compression efficiency is defined as the ratio of the work required for an isentropic compression process to the actual work required. Few data are available.

additional capacity will manifest itself by lowering the evaporating temperature t_3 in equation 12.5 with increased capacity of the evaporator and appropriate adjustments of the other factors.

12.12. A Condensing Unit consists of a compressor and motor and either an air-cooled finned condenser or shell and tube water-cooled condenser (Fig. 12.8). Rating in Btu per hr at various evaporator temperatures and ambient air temperature or condensing water temperatures is given. Since most compressors are belt driven, capacity can be further adjusted by change of compressor speed.

Equations 12.6 and 12.7 are useful for estimating power requirements for conditions not covered by commercial data.

CONTROLS

Refrigeration control may be considered from the standpoint of (1) individual component control, (2) safety of both equipment and operator, and (3) temperature and perhaps humidity control of the medium being cooled. Temperature and humidity will be considered.

12.13. Motor Circuit Thermostats are used on single-evaporator systems with capillary tube, thermal expansion valve, and float-controlled evaporators. Temperature control is as accurate as the thermostat. Expansion or float valves that do not completely restrict liquid flow during the off-cycle may cause compressor flooding. Compressor damage or motor overload may result when the unit starts.

12.14. A Low-Side-Pressure Switch opens and closes the motor circuit on the basis of the low side-pressure which is directly related to the temperature of the evaporating liquid as shown in equation 12.5. As cooling takes place, t_3 and the corresponding refrigerant saturation pressure drops. The decreased pressure opens the switch. The residual liquid refrigerant soon assumes ambient temperature, the pressure rising to the corresponding saturation pressure. A rise in ambient temperature causes a pressure rise that closes the circuit. Expansion and compressor valve leakage are compensated for by automatic compressor operation. Frost accumulations decrease the U valve of the evaporator, and the controlled temperature rises. The controlled temperature rises as the refrigerating rate increases

because the evaporating-refrigerant and cooling-material temperature difference increases. This method of control is satisfactory only for single-evaporator systems unless additional control features such as those discussed below are added to the system.

12.15. Magnetic Valves operated by a thermostat are frequently used for control in multiple-evaporator systems, Fig.

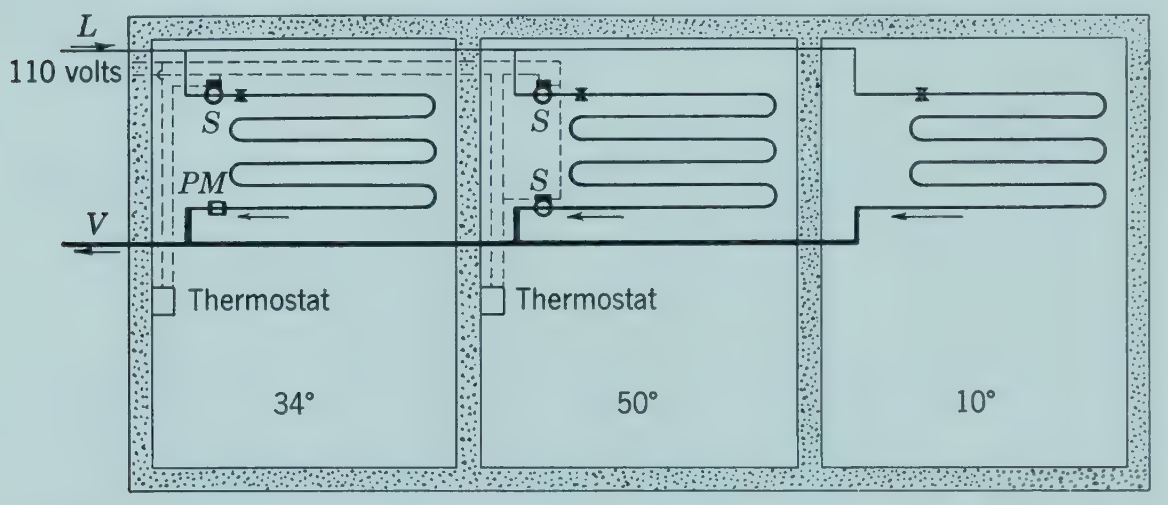


Fig. 12.9. Multiple-evaporator system operating from a single compressor. The compressor motor is operated by a low-side-pressure switch activated by the low side pressure of the lowest temperature load, 10°F in this example. The solenoids of the higher-temperature loads are controlled by thermostats. Liquid refrigerant is let into the evaporator only when cooling is required. The evaporative pressure-maintaining valve *PM* holds the pressure in the evaporator of the 34°F load above the pressure in the vapor return line. This permits humidity control or minimum evaporator frosting by controlling the evaporating temperature. The evaporating temperature of the 50°F load is the same as that of the 10°F load. A vapor discharge solenoid can be used as installed on the 50°F load to provide more accurate temperature control. This valve confines the residual liquid refrigerant in the evaporator, thus stopping the refrigerating action at the exact prescribed temperature. Without this valve the residual liquid would evaporate and the temperature might drop below the controlled temperature. This valve is used only where highly exact control is required.

12.9. The thermostat opens the valve permitting liquid to enter the evaporator. The suction pressure switch is set for a pressure somewhat below the operating pressure of the evaporator. Magnetic valves are also used in combination with manual and float expansion valves to facilitate operation.

12.16. Evaporative-Pressure-Maintaining Valves, also called back-pressure control valves, can be used on multiple-evaporator systems to control the evaporating pressure in any evaporator.

Without them, the evaporating pressure and temperature of all the evaporators in a multiple system are those of the lowest pressure unit with slight variations for line friction. If a zero-degree room and a 34-degree room are to operate on the same compressor it will probably be advisable to operate the higher temperature room with a higher pressure evaporator in order to minimize coil frosting and, if desired, to maintain a high relative humidity in the room. This is done by using a pressure-maintaining valve shown in principle in Fig. 12.10 and located in Fig. 12.9.

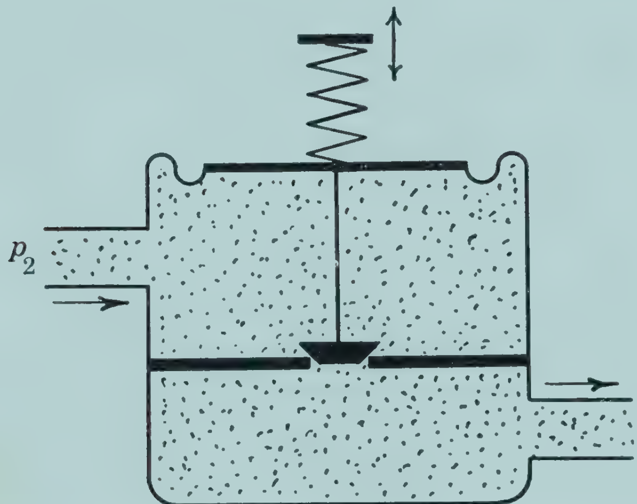


Fig. 12.10. Schematic sketch of an evaporative pressure-maintaining valve.

The thermodynamics of the process are shown in Fig. 12.11. The flow through the valve is irreversible adiabatic, state change $a_2 - a'$. The power state path for the higher pressure unit would be $a_2 - b_2$ if separate compressors were used for each load, the power energy for the system being

$$W_2(h_{b_2} - h_{a_2}) + W_1(h_{b_1} - h_{a_1}) \tag{12.8}$$

W_2 and W_1 are the respective refrigerant weight rates. The total power for the system designed for a pressure-maintaining valve and a single compressor is

$$(W_2 + W_1)(h_{b_3} - h_{a_3}) \tag{12.9}$$

The difference between equations 12.8 and 12.9 is the extra power required owing to the pressure-maintaining valve.

Multiple-evaporator systems with an operating temperature range and pressure-maintaining valves on the high-temperature evaporators are economically sound if the major portion of the

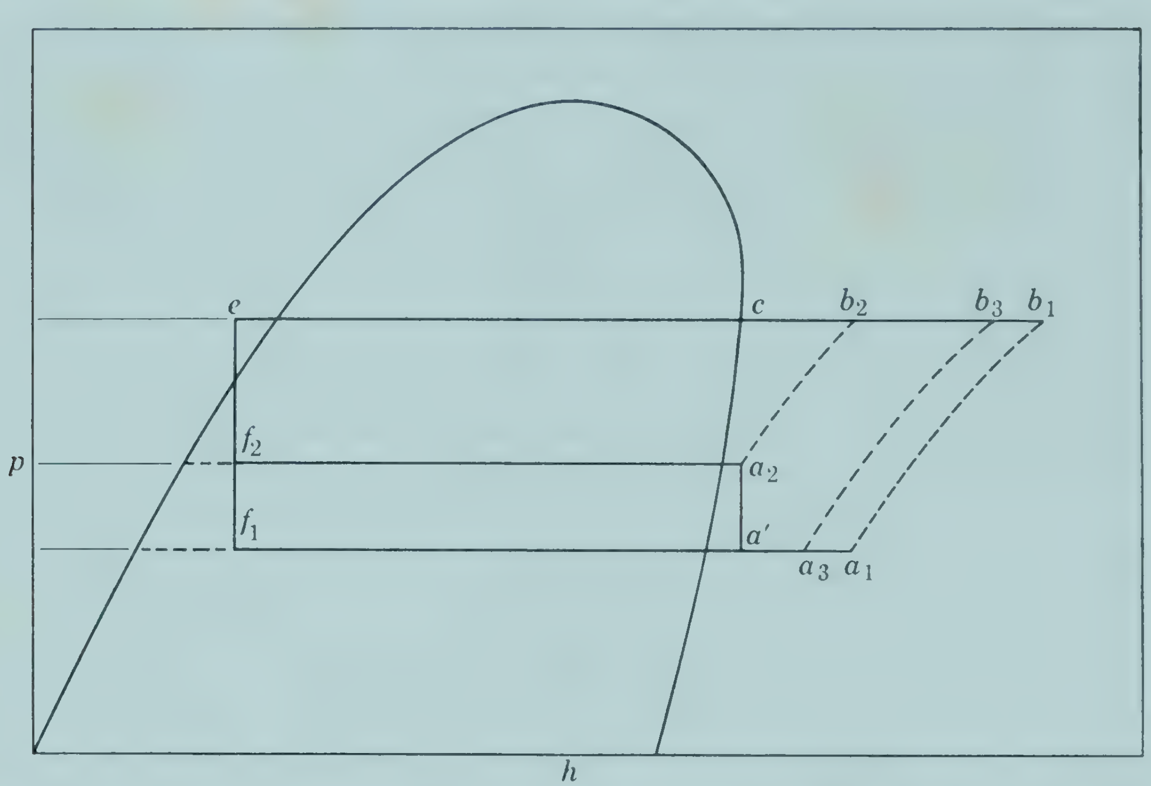


Fig. 12.11. The thermodynamic process of the evaporative pressure maintaining valve and its relation to the entire system.

load is a low-temperature load. If the major portion of the load is high temperature, independent systems may be advisable.

HEAT PUMP

The heat pump is a refrigeration machine installed where the heat discharged from the condenser is desired rather than the heat absorbed by the evaporator. The heat pump may be regarded as a device that lifts or "pumps" heat energy from a low-temperature source for use at a higher temperature. It is used satisfactorily for space heating by lifting heat from the ground, outside air, or bodies of water at a lower temperature than the space to be heated. Other uses, existing and proposed, make use of both evaporator and condenser energies.

Heat pumps may be fitted to a variety of operations, e.g.,

1. Comfort heating and cooling of buildings.
2. Water and other liquid heating, domestic and industrial.
3. Evaporating, concentrating, and distilling.
4. Drying.
5. Simultaneous heating and cooling, e.g.,

a. Water cooling and space heating.
b. Heating and dehumidifying, domestic and industrial.
c. Water heating and space cooling.

12.17. Coefficient of Performance. The coefficient of performance of a heat pump refers high-temperature condenser energy to the driving energy so that the cycle c.o.p. is (referring to Fig. 12.2)

$$(h_{c'} - h_d)/(h_b - h_a) \tag{12.10}$$

the Carnot c.o.p. is

$$T_H/(T_H - T_C) \tag{12.11}$$

Since the c.o.p. relates the useful heat output as refrigeration in a refrigerator or heat energy input in a heat pump, the c.o.p. for simultaneous usage such as 5*a* could be considered as

$$[(h_{g'} - h_f) + (h_{c'} - h_d)]/(h_b - h_a) \tag{12.12}$$

The smaller the difference in temperature between the evaporator and condenser, the greater the c.o.p. and the greater will be the output per unit of mechanical input.

A decision to install a heat pump in lieu of a conventional gas, oil, or other heat energy source should be made on the basis of an economic and convenience study. Factors that must be considered are initial cost, length of life, upkeep, operating cost and continuity of power source, operating attention, etc. For example, a gas-fired hot-water heater might be 70 per cent efficient thermally; thus, 1400 Btu must be supplied for each 1000 Btu taken up by the water. A heat pump with a c.o.p. of 3.5 would need only mechanical energy the equivalent of 290 Btu.

SOURCES AND SINKS

The "source" of heat for a heat pump can be a steady-state source such as air or water. The mass rate would be controlled so that the difference in temperature between the evaporating refrigerant and the cooling medium is nearly constant for various heat rates. Air, well water, stream water, liquid manufacturing wastes, etc., can be used.

Systems installed in areas where air and water temperatures are low or where a sufficient supply of water or other heat-source

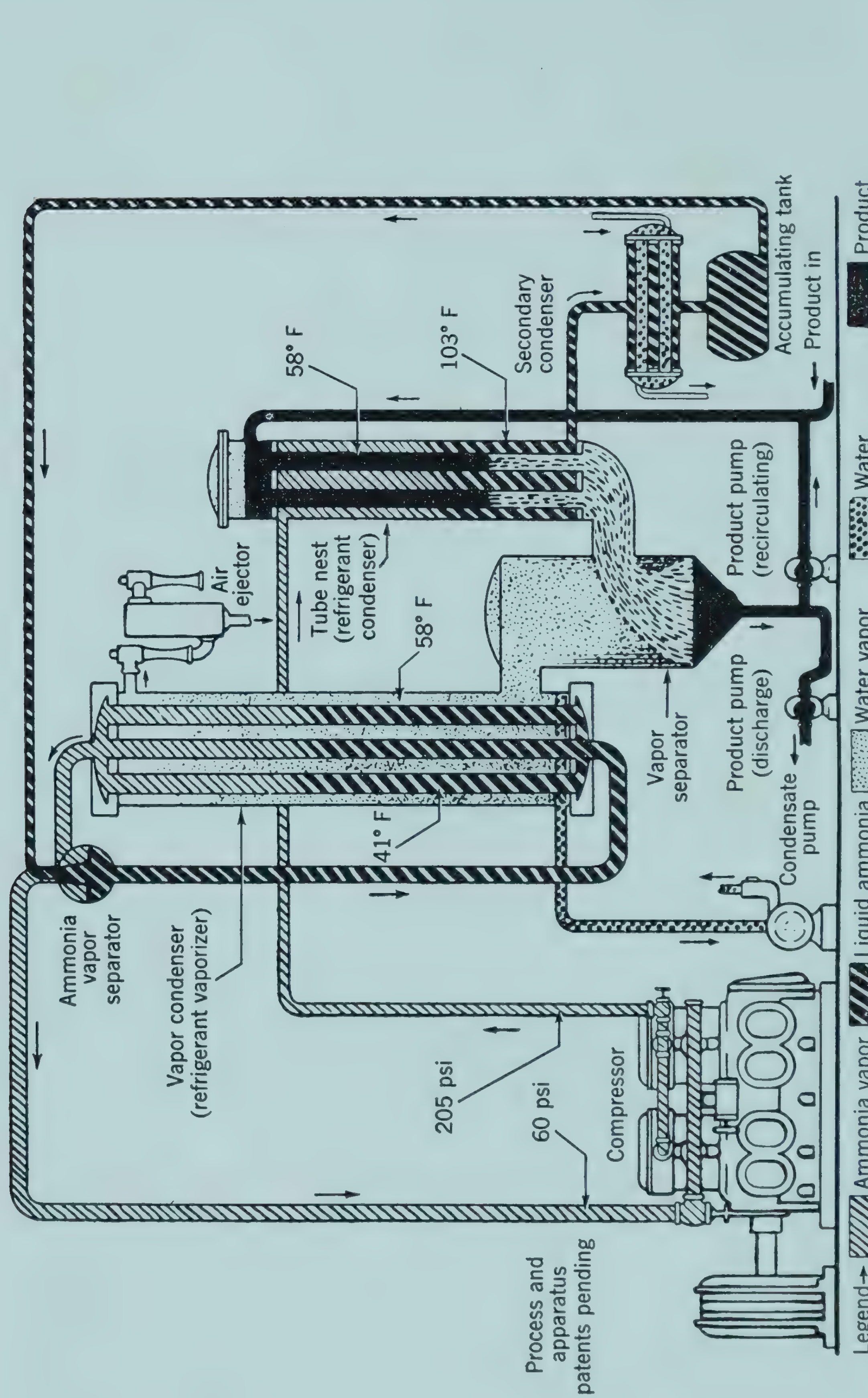


Fig. 12.12. A heat pump used for concentrating orange juice. (*Courtesy* the Mojonnier Co.)

medium is unavailable may require a transient source. A transient source is a pond, well, or series of wells, the earth, or other stationary heat source with sufficient capacity and transfer properties to supply the heat at a satisfactory rate. Poor performance may occur if, when designing the source, decrease in heat transfer due to shallow thermal gradients as operation continues is not recognized.

A "sink" is a heat disposal for condenser heat. The same characteristics apply as for the source.

A heat pump used for concentrating orange juice is an example of an agricultural heat-pump installation. Fig. 12.12 shows a schematic arrangement of the system and the operating conditions. The unique feature is the use of the refrigeration evaporator for condensing the vapor removed from the orange juice. The compressor is an 11 by 10 in. 4-cylinder 300-rpm unit with a volumetric efficiency of 91 per cent. The orange juice is concentrated from 11 to 55 degrees Brix (specific gravity of 1.03 to 1.16).

Note that the power required is nearly an inverse function of the heat-exchange area. If the heat-exchange areas were increased the compressor power could be reduced. The optimum size of these factors must be based upon a cost analysis.

REFERENCES

1. Macintire, H. J., and Hutchinson, F. W. *Refrigeration Engineering.* Second Edition. Wiley. 1950.
2. Myers, W. P., Jr. Refrigeration Controls. *Refrig. Eng.* 57:884ff. Sept., 1949.
3. Penrod, Estel B. Development of the Heat Pump. *Ky. Eng. Expt. Sta. Bull. 1,* 4. 1947.
4. Raber, B. F., and Hutchinson, F. W. *Refrigeration and Air Conditioning Engineering.* Wiley. 1945.
5. *The Refrigerating Data Book. Basic Volume.* Sixth Edition. The American Society of Refrigerating Engineers. 1949.
6. *The Refrigerating Data Book. Refrigeration Applications Volume.* Third Edition. The American Society of Refrigeration Engineers. 1951.

PROBLEMS

1. Ammonia is required for a 3-ton refrigeration system with an evaporating temperature of $-10°F$. The condenser will be water-cooled and

will operate on a 10°F differential. Tap water temperature is 70°F. Determine the following. Assume saturated conditions.

a. High and low side-pressures (gage).
b. Compressor displacement rate; assume volumetric efficiency 85 per cent.
c. Thermal horsepower required, assume thermal efficiency 90 per cent.
d. The liquid ammonia rate in pints per minute.
e. The capacity in tons if the evaporating temperature is raised to 15°F. The compressor capacity is unchanged.

2. Work problem 1 with Freon-12 as the refrigerant. Compare compressor displacement and liquid rate. Which refrigerant would you recommend for a large installation? Why?
3. A 1-ton Freon-12 system has an evaporator temperature of 32°F and a high side-pressure of 90 psig. Liquid refrigerant enters the evaporator at 60°F. Vapor enters the compressor at 50°F. Use the Mollier chart and determine:

a. Per cent flashing into vapor as the liquid passes through the expansion valve. Calculate from enthalpy values.
b. The coefficient of performance.
c. The thermal horsepower.
d. The thermal horsepower if the vapor enters the compressor with no superheat.
e. The compressor displacement, assume 80 per cent efficient.

4. An evaporator pressure-maintaining valve holds a 30 psig evaporating pressure against a 5 psig low side-pressure. The vapor enters the valve saturated. Determine the energy in ft lb per lb of refrigerant expended in the process.
5. Show by a Mollier chart the necessity for the secondary condenser of Fig. 12.12.
6. Determine the coefficient of performance of the heat pump of Fig. 12.12.

CHAPTER 13

Process Condition Observations, Records, and Controls

NOMENCLATURE

A = surface area of sensing element, sq ft.
C = heat capacity of sensing element, Btu per °F.
E_g = voltage generated by thermocouple.
E_m = voltage at instrument.
h = unit surface thermal conductance, Btu per (°F ft^2 hr).
I = current, amp.
R_c = resistance of external circuit, ohms.
R_m = resistance of instrument, ohms.
t = temperature of sensing element, °F.
t_a = temperature of adjacent medium, °F.
t_{av} = average air temperature, °F.
t_p = amplitude, °F.
δ = lag of sensing element in periodic environment, radians.
θ = time, hr.
θ_p = period of wave, hr.

The operating conditions of a process frequently must be controlled within specific finite limits. Cold storage and freezing, pasteurization, homogenization, washing, drying, and dehydration are processes that require temperature, pressure, flow rates, etc., to be controlled. Temperature is the most frequently controlled condition. Although this chapter deals with temperature in the main, adequate treatment of some other factors is included.

OBSERVATIONS, TEMPERATURE

13.1. Liquid-in-Glass Thermometers. Liquid-in-glass thermometers are suitable for observing temperatures up to approximately 950°F. Mercury is used as the actuating fluid for temperatures between −35° and 950°F. Mercury cannot be used

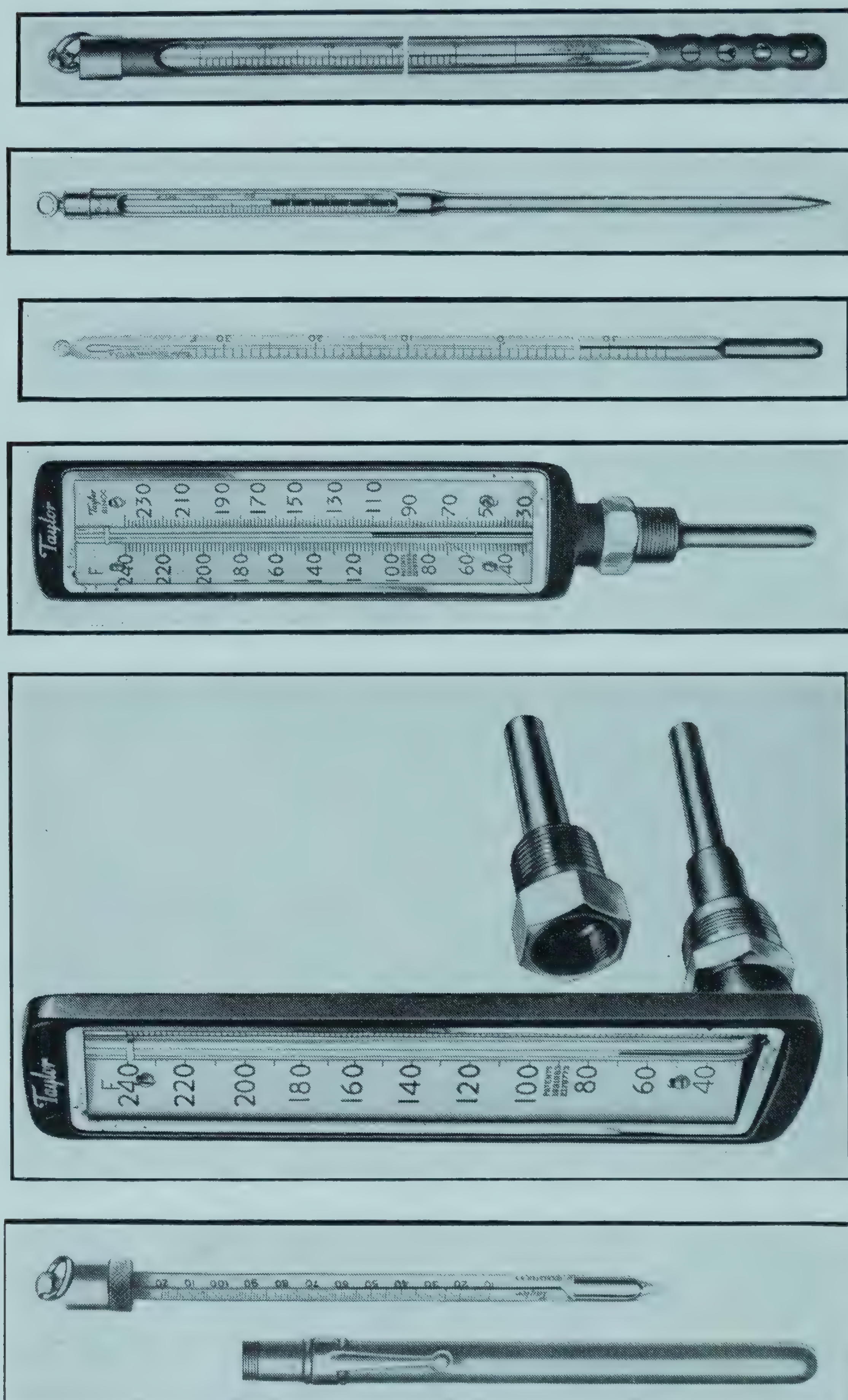


Fig. 13.1. A few of the many forms of **liquid-in-glass** thermometers. (*Courtesy* the Taylor Instrument Co.)

for temperatures below its freezing point of $-38°F$. Alcohol or other organic fluids are used for atmospheric temperatures and for temperatures down to $-200°F$.

The accuracy is usually 1 per cent of the range when properly installed and operated. Individually calibrated thermometers with a high degree of accuracy can be secured. The amount of immersion and conditions surrounding the exposed portion of the thermometer affect accuracy. The amount of immersion is usually specified by the manufacturer. Widely fluctuating ambient temperatures in the exposed portion may cause observational errors due to expansion and contraction of the glass tube.

Glass thermometers are available in a number of forms; some are shown in Fig. 13.1.

13.2. Bimetallic Thermometers. Bimetallic thermometers can be made comparable to mercury glass thermometers in accuracy, temperature range, and uses.

Two strips of dissimilar metal, one generally being Invar which has a very low coefficient of thermal expansion, are welded or fused together as shown in Fig. 13.2. Changes in temperature cause movement of the free end which can be linked to an indicating needle. The bimetal strip can be straight or coiled, Fig. 13.2.

This type of thermometer is superior to the glass thermometer in that the bimetallic thermometer is more rugged, easier to read, and is not affected adversely by ambient temperatures. The bimetallic unit must be immersed completely. Since the relative motion is essentially linear with temperature, calibration is easy and simple motion-transfer links can be used.

13.3. Pressure Thermometers. A pressure thermometer consists of a sensing bulb and a Bourdon tube, bellows, or a pressure spring connected by a capillary tube. A temperature change at the bulb causes a change in pressure within the system. The resulting movement of the pointer linked to the pressure spring indicates the temperature at the bulb. Three types of systems are used.

1. *Mercury-Filled Systems.* Steel or stainless-steel systems are completely filled under high pressure with mercury, which has a much greater coefficient of thermal expansion than the steel. Consequently, a change of temperature will cause a relative volume

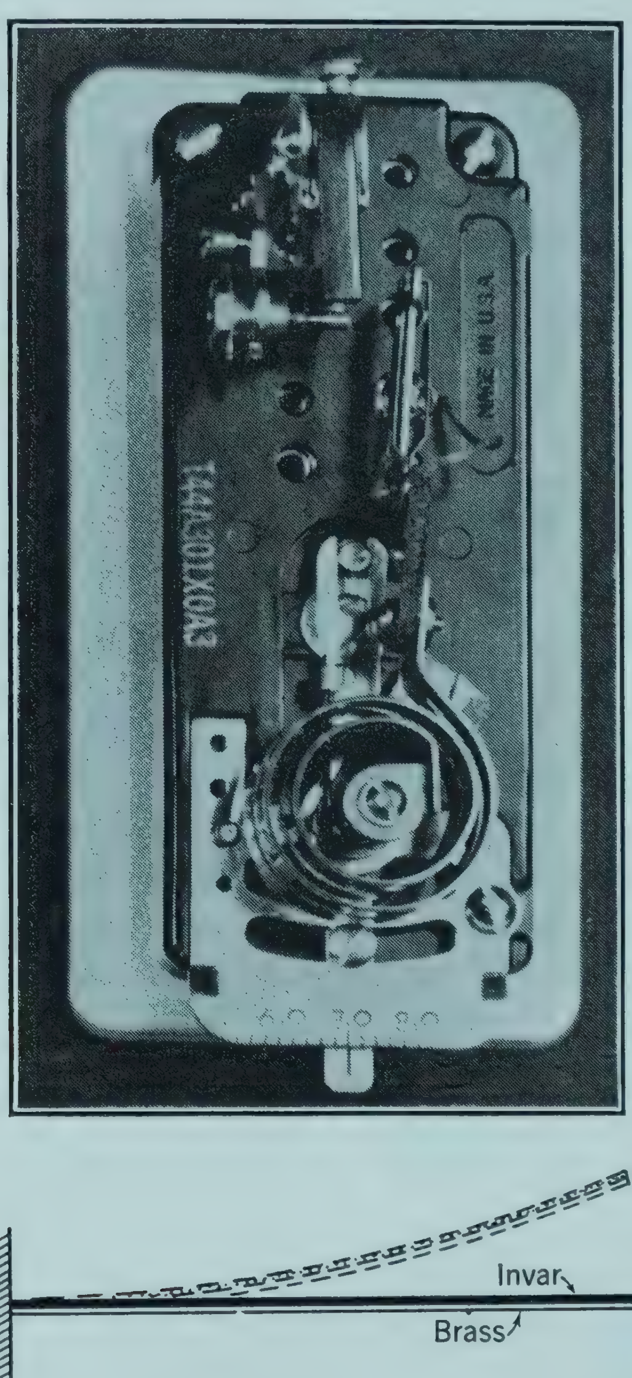


Fig. 13.2. A bimetal thermometer. When the temperature increases, the brass expands more than the Invar and it takes the shape shown dotted. The bimetal units are used in thermometers and controls. An on-off temperature control is shown above. (*Courtesy* Minneapolis-Honeywell Regulator Co.)

change which causes the pressure spring to move. The temperature range is the same as that for mercury in glass thermometers, namely, $-35°$ to $950°F$. Temperature response is nearly linear through the operating range of a specific instrument. The bulb may be located up to 200 ft away from the indicator, the distance being limited by cost more than performance.

Variation of the pressure spring and connecting capillary temperature, with respect to that of the bulb, may produce a significant observational error. Compensators are used to correct for ambient temperature effects which are produced in this manner.

A bimetal link in the mechanism or a complete pressure-spring system opposing the motion of the main spring will correct for a

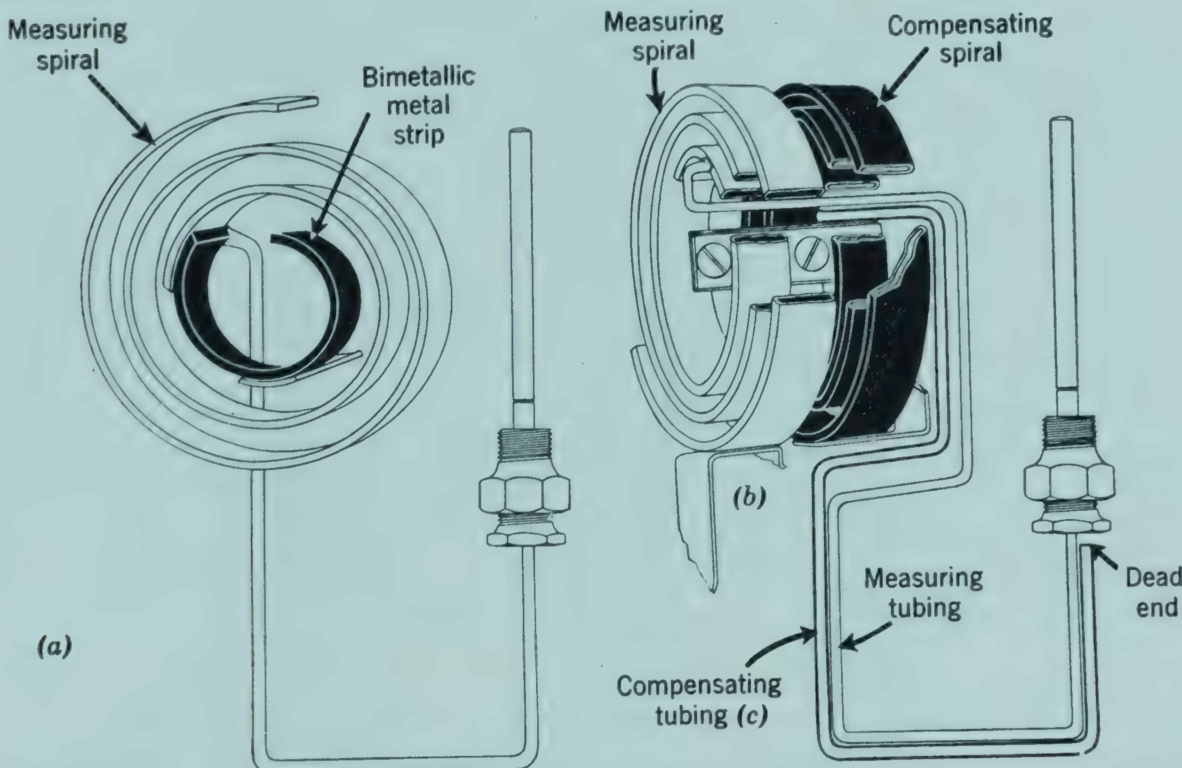


Fig. 13.3. Two systems used to correct for ambient temperature error in pressure-spring-thermometer systems. (*Courtesy* Minneapolis-Honeywell Regulator Co.)

temperature variation of the indicating mechanism as shown in Fig. 13.3*a*. When the capillary tube is short, the volume of mercury in the tube is small when compared to the volume in the bulb and pressure spring; here compensation for change in capillary fluid volume due to ambient temperature change may not be necessary. If the tube is long, ambient temperature errors are corrected by using a blanked capillary tube attached to the compensating pressure spring, Fig. 13.3*c*. Thus, temperature variation at any point is completely corrected for by counter movement of the compensating system. Compensation is also brought about by inserting an Invar wire inside the capillary tube. The dimensions are so selected that the change in volume of the Invar wire

due to a change in temperature equals the change in capillary mercury volume.

A change in elevation of the bulb relative to the pressure spring will cause a shift in the pressure reading equal to the elevation head. A simple adjustment is provided to readjust the instrument after installation in a particular location.

Oil and other liquids are also used in liquid-filled instruments.

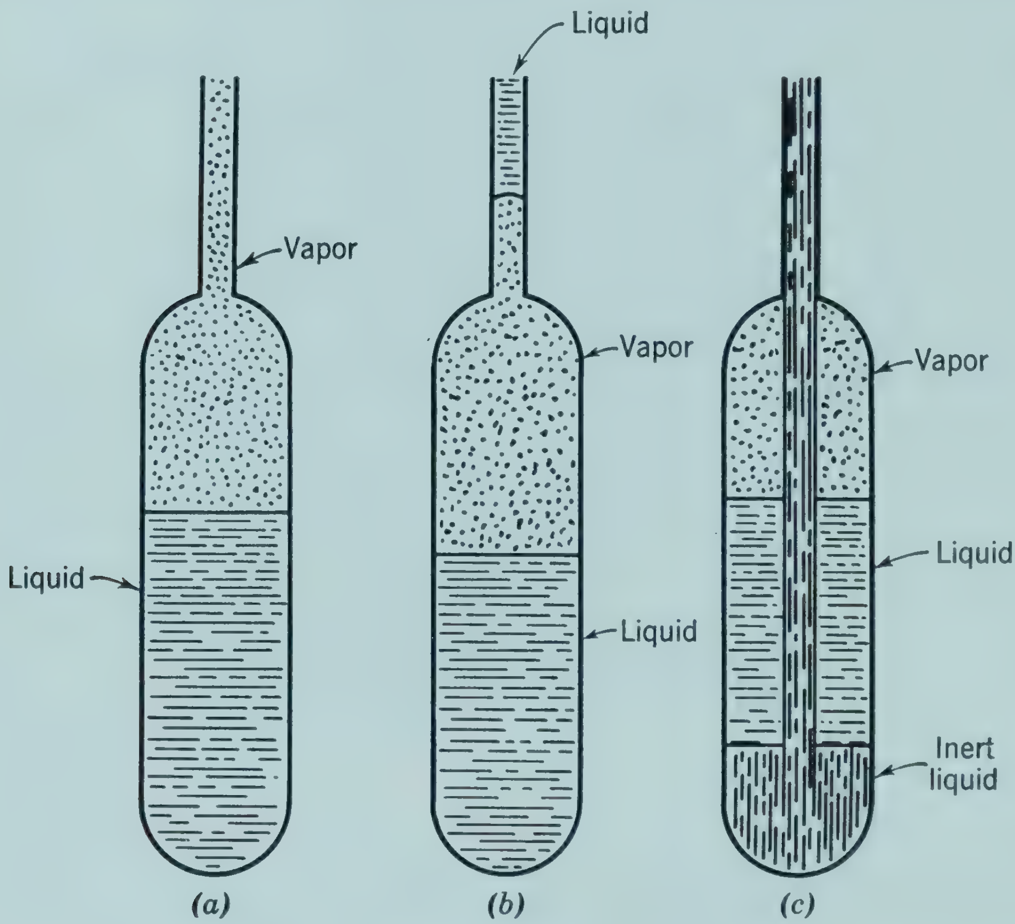


Fig. 13.4. Types of fill for vapor-pressure temperature-indicating systems. The inert liquid must not react with the vaporizing liquid or produce a significant vapor pressure at the operating temperatures.

2. *Gas-Filled Systems.* Systems filled with high-pressure gas, usually nitrogen, perform according to the gas law so that the system volume and resultant mechanism movement are nearly proportional to the change in temperature. Compensation for ambient temperature variation can be brought about in the same manner as with mercury-filled systems. However, larger sensing bulbs are required with gas than with mercury because of the compressibility of a gas.

Nitrogen-filled systems are designed for operation between $-200°$ and $800°F$.

3. *Vapor-Pressure Systems.* Some systems are powered with a volatile liquid as shown in Fig. 13.4. The pressure at the indicating spring is the saturated pressure of the fluid at the temperature of the bulb. Ambient temperature compensation is not required if the system is properly designed so that the liquid-vapor interface is at the required temperature. The response is not proportional to temperature since the saturated vapor pressure of liquids is nonlinear with temperature. Therefore, the temperature scale expands at higher temperatures.

The bulb is filled in one of the three ways shown in Fig. 13.4. Fill *A* must always be specified and used when the spring and capillary temperatures are higher than the bulb temperature. Thus the vapor is superheated and a true bulb saturated pressure is effective upon the spring. If the temperature of the coil or any part of the capillary drops below the temperature of the bulb, vapor will condense at the lower temperature which will then be indicated by the instrument. Fill *B* must obviously be used at temperatures that are always higher than the temperature of spring and capillary. Fill *C* must be used if ambient temperatures fluctuate above and below bulb temperatures.

The temperature range for this type of fill is approximately $-20°$ to $600°F$.

13.4. Thermocouple Thermometers. When two wires of dissimilar metals are joined in a circuit as shown in Fig. 13.5 and

Fig. 13.5. An elementary thermocouple thermometer. The measuring instrument is a milliammeter or a galvanometer.

the reference junction t_r and the measuring junction t_m are at different temperatures, a difference in voltage occurs between the junctions. This difference in voltage can be used to determine the difference in temperature, by reference to tables based on the established properties of the metals. Or, in a circuit of known resistance, a deflection galvanometer can be used to measure the

current, for which the voltage, and thus the temperature difference, is found.

The common wire combinations and their important characteristics are listed in Table 13.1.

Table 13.1 THERMOCOUPLE WIRE COMBINATIONS

Couple	*Composition*	*Approximate Useful Temperature Range, °F*	*Approximate Response, mv per °F*
Copper–Constantan	100 Cu–55 Cu, 44 Ni	−300 to 700	0.0232
Iron–Constantan	100 Fe–55 Cu, 44 Ni	0 to 1400	0.0292
Chromel–Alumel	90 Ni, 9 Cr–97 Ni, 3 Al	600 to 2200	0.0231
Platinum–platinum, 13% rhodium	100 Pt–87 Pt, 13 Rh	1300 to 3000	0.00761
Platinum–platinum, 10% rhodium	100 Pt–90 Pt, 10 Rh	1300 to 3000	0.00660

For higher accuracy, a null method using a potentiometer is preferred. A standard voltage is produced on the slide wire shown in Fig. 13.6. The contact point on the rheostat is then moved

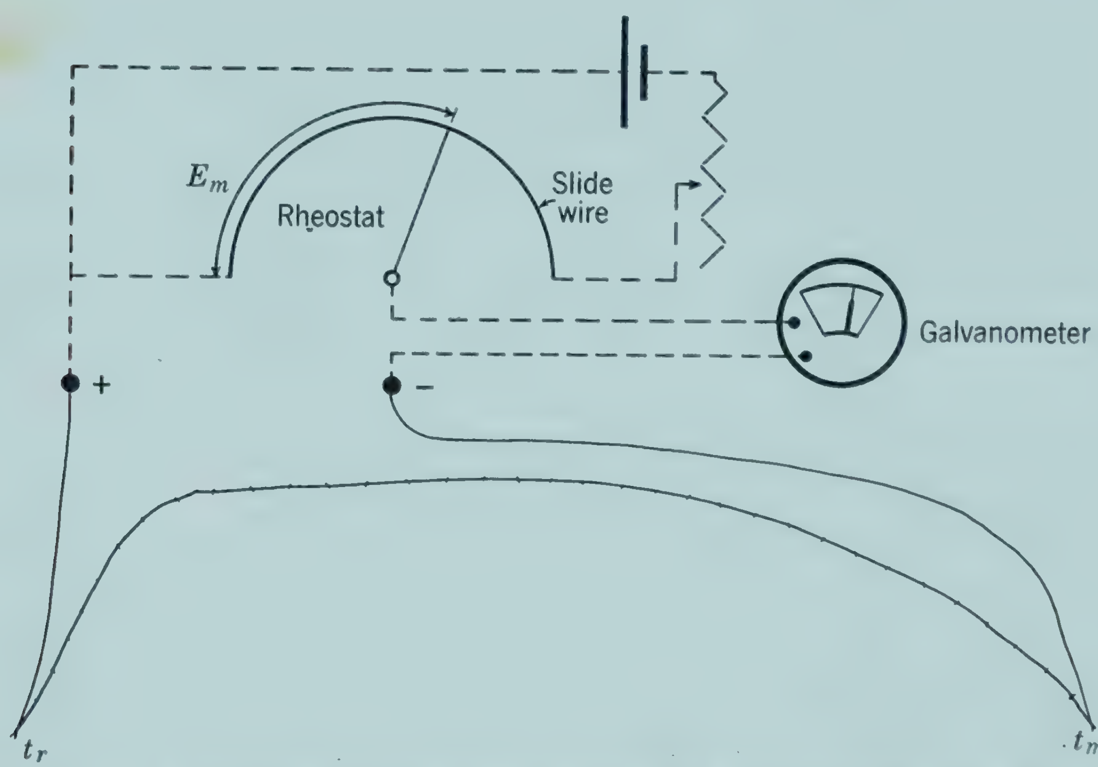


Fig. 13.6. A potentiometer used for reading temperatures with thermocouples.

until the galvanometer shows a null reading, thus indicating that no current is flowing in the external circuit and that the voltage resulting from $t_m - t_r$ is equal and opposing that across the portion of the slide wire E_m. Consequently, the temperature differ-

ence $t_m - t_r$ can be represented by the position of the contact point. Since the reading is made when no current is flowing through the thermocouple leads, standardization of the length and size of the wires and type of junctions, i.e., of the resistance of the external circuit, is unnecessary.

The indicator of a pyrometer is graduated in milliamperes or directly in degrees if a standard reference temperature is used. The potentiometer is graduated in millivolts or in degrees if a standard reference temperature is used. A standard reference temperature is a fixed temperature such as 32°F. Commercial instruments are frequently fitted with compensating devices, which automatically adjust the circuit for variation of t_r and thus eliminate the need for consideration of t_r when using the instrument. The accuracy or reproducibility depends upon the consistency of the wire composition. Usual accuracy in terms of degrees per millivolt is ½ to 1 per cent. All the couples can be used satisfactorily down to about −400°F. The response of platinum couples at low temperature difference is too small for acceptable use. The temperature-millivolt relationship is not quite linear for any couple. Therefore, calibration data appropriate to the temperature range and reference temperature must be used.

Deflection instruments are simpler but less accurate than null instruments because of possible variations in circuit resistance. By Ohm's law, the current in Fig. 13.5 is

$$I = E_g/(R_c + R_m) \tag{13.1}$$

From Equation 13.1

$$E_m/E_g = R_m/(R_m + R_c) \tag{13.2}$$

where I = current, amp.
E_g = voltage generated by thermocouple.
E_m = voltage at instrument.
R_c = resistance of external circuit, i.e., couple and leads, ohms.
R_m = resistance of instrument, ohms.

Thus, by equation 13.2, the higher the resistance of the instrument, and the lower the circuit resistance, the closer the measured voltage is to the voltage generated by the couple. A given in-

strument can of course be calibrated for use with a known circuit, but the error due to variations in circuit resistance will be minimized with a high ratio of instrument to circuit resistance.

Instrument leads should preferably be of the same composition as one of the thermocouple metals. When they are not, particular care must be taken to avoid having the couple-to-lead connections differ in temperature, else local couple-to-lead thermoelectric voltages will introduce errors.

Deflection-instrument thermocouple indicators employed for high-temperature observations are often called pyrometers. Platinum-platinum-alloy thermocouples are preferred for pyrometry because they are more resistant to oxidation at the high temperatures.

INSTRUMENT RESPONSE

Instruments must have not only specified accuracy but also suitable rate of response to change in measured variable. When, for example, the temperature environment of a thermometer drops, the instrument does not instantly indicate the new temperature. As heat flows from the bulb to the surroundings, its temperature falls, thus

$$C\,dt = -hA(t - t_a)\,d\theta \tag{13.3}$$

where C = heat capacity of sensing element, Btu per °F.
t = temperature of sensing element, °F.
h = unit surface thermal conductance (heat-transfer coefficient of sensing element) Btu per (°F sq ft hr).
A = surface area of sensing element, sq ft.
t_a = temperature of the adjacent medium, °F.
θ = time, hr.

For the case where the surrounding medium is suddenly changed from t_0 to t_a at time $\theta = 0$ and then held constant, equation 13.3 integrates to give

$$(t - t_a)/(t_0 - t_a) = e^{(-hA/C)\theta} \tag{13.4}$$

as illustrated in Fig. 13.7*a*.

Example. A thermometer bulb with a heat capacity of 0.024 Btu per °F and a surface area of 0.06 sq ft is at 70°F. It is suddenly placed in surroundings at 30°F. If the surface heat-transfer coefficient is 2 Btu per (°F

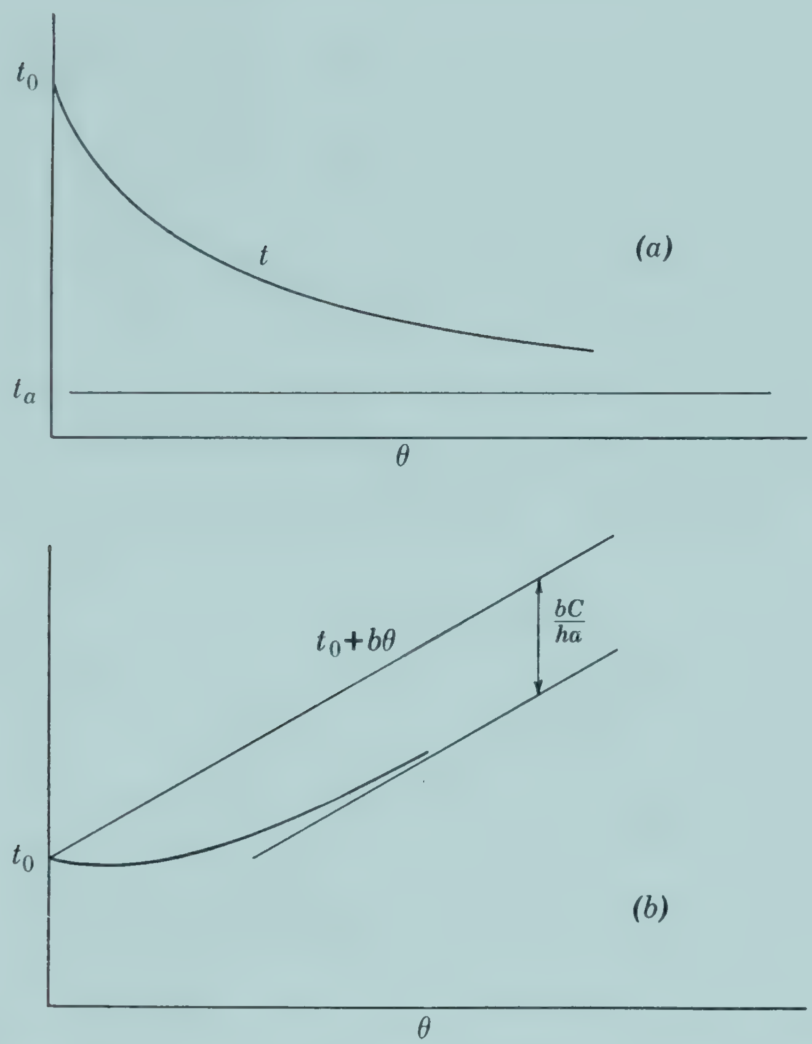


Fig. 13.7. Instrument response to a sudden temperature change (*a*) and to a linear temperature change (*b*).

sq ft hr), what time will be required for the bulb to be within 2°F of the new surroundings?

$$(32 - 30)/(70 - 30) = e^{-(2 \times 0.06/0.024)\theta}$$

$$\theta = -1/5.00 \ln 0.05 = 0.6 \text{ hr} \quad \text{or} \quad 36 \text{ min}$$

13.5. Response to Linear Change in Air Temperature. In this case, the bulb and surroundings are initially at t_0. The surroundings suddenly start to rise at b degrees per hour, thus

$$t_a = t_0 + b\theta \tag{13.5}$$

Substitution of t_a from equation 13.5 into equation 13.3 and integration yields

$$t = t_0 + b\theta - b(C/hA)(1 - e^{-(hA/C)\theta}) \tag{13.6}$$

Equation 13.6 is illustrated in Fig. 13.7*b*. As time goes on, the exponential term approaches zero. The bulb then differs from

the surroundings by bC/hA degrees. The error thus depends directly upon the rate of rise and heat capacity of the bulb and inversely upon the surface conductance. The measured temperature lags behind the temperature of the surroundings by C/hA hr.

Example. The bulb of the thermometer in the example above is located in surroundings that change at the rate of 30°F per hr. Find the error at ½ hr after the change is initiated, if the surface heat-transfer coefficient is 2 Btu per (°F sq ft hr).

The error after the exponential term has become negligible is bC/hA or $30 \times 0.024/2 \times 0.06$ which is 6°F. However, at ½ hr, the exponential term is $e^{-\frac{2 \times 0.06}{0.024} \times 0.5}$ or 0.082. The error is then $6(1 - 0.082)$ or 5.4°F.

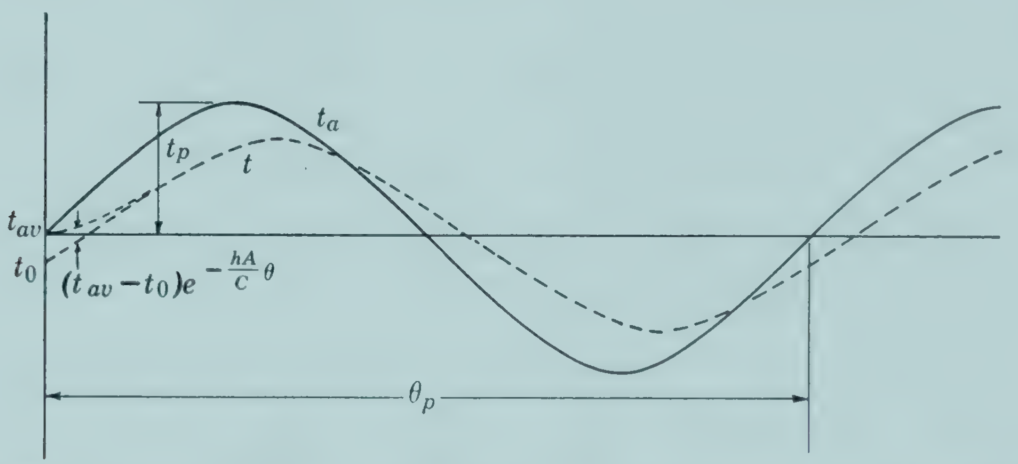


Fig. 13.8. Response of an instrument to a sinusoidal temperature change

13.6. Response to Sinusoidal Air-Temperature Change. When the surroundings of a thermometer change sinusoidally as shown in Fig. 13.8,

$$t_a = t_{\text{av}} + t_p \sin\left(\frac{2\pi\theta}{\theta_p}\right) \tag{13.7}$$

where t_{av} = average air temperature, °F.
t_p = amplitude, or half-range of the sinusoidal variation, °F.
θ_p = period of the wave, hr.

The differential equation (equation 13.3 with t_a substituted from equation 13.7) yields on integration

$$t = t_{\text{av}} + \frac{t_p}{\sqrt{1 + \left(\frac{2\pi C}{\theta_p hA}\right)^2}} \sin\left(\frac{2\pi\theta}{\theta_p} - \delta\right) + Ke^{-(hA/C)\theta} \tag{13.8}$$

in which $\delta = \tan^{-1}(2\pi C/\theta_p hA)$ radians.

K = a constant of integration, which is evaluated by substituting values of t and t_a when $\theta = 0$.

The last term in Equation 13.8 dies out as time progresses. After this, equation 13.8 resembles equation 13.7, except for the decrease in amplitude and the lag. The amplitude ratio $1/\sqrt{1 + (2\pi C/\theta_p hA)^2}$ is usually of more concern than the lag. The lag, which is expressed as a time angle δ is converted to hours by multiplying by $\theta_p/2\pi$.

Periodic temperatures that are not simple sine waves can be expressed as a Fourier series, and the response to each harmonic can be found by equation 13.8.

Example. The bulb of the previous examples is used to measure a sinusoidal temperature having a period of ½ hr and a range of 10°F ($t_p = 5$). Predict the range that the thermometer will indicate and also the lag.

From equation 13.8, the amplitude after a few cycles will be

$$\frac{5}{\sqrt{1 + \left(\frac{2\pi \times 0.024}{\frac{1}{2} \times 2 \times 0.06}\right)^2}} = \frac{5}{1 + 2.51^2} = \frac{5}{2.7} = 1.85°\text{F}$$

The range, twice the amplitude, is 3.7°F. It is obvious that this thermometer represents the measured variable very poorly. Its performance would be improved by directing an air blast over the bulb to increase the heat-transfer coefficient.

The lag is $\tan^{-1}(2\pi \times 0.024/\frac{1}{2} \times 2 \times 0.06)$, which is 68.5° or 1.2 radians. The lag in hours is $(\frac{1}{2}/2\pi) \times 1.2$ or 0.096 hr.

13.7. Response Summary. The response characteristics discussed in the previous sections are related to the heat-transfer features of the sensing element in the expression

$$C/hA$$

This has the dimension of time and is often called the time constant of the thermometer in the given system.

Observations are most accurate when the value of this expression is small. Therefore, the specific heat, specific weight, and volume of the sensing unit should be small and the surface heat-transfer coefficient and area large if fast response is desired.

Thus, long narrow or coiled sensing units are preferred to short thick units. The volume of the unit should be as small as practical. Thermocouples made of small wire, 36 gage for example,

are highly acceptable in this respect. The surface heat-transfer coefficient is high for liquids or condensing vapors and instrument response is fast. Bulb-type instruments to be used with gases should be gas filled if fast response is desired.

OBSERVATIONS, PRESSURE

The devices and techniques available for measuring pressures are discussed in Chap. 2. Coil springs and bellows similar to those used for temperature observations are also used for pressure observations. These elements perform in a manner comparable to the Bourdon tube, sect. 3.4.

Since the gage tube is usually connected directly, or with a simple valve, to the vessel or pipe whose pressure is to be measured, the resistance to transfer of energy here is negligible, so that a response analysis similar to that developed for the thermometer is not required. Where frequency of pulsation of pressure approaches the natural frequency of the gage mechanism as a spring-mass system, serious vibration occurs. This can be reduced by throttling with the gage valve. A routine schedule of checking the gage response should be adopted in such a case, to avoid having a gage become inactive from plugging of the nearly closed valve.

OBSERVATIONS, RELATIVE HUMIDITY

Relative humidity, defined and discussed in Chap. 10, can be observed by four methods.

13.8. Wet-Bulb Psychrometer. The wet-bulb method of observing relative humidity is discussed in sects. 10.6 and 10.11. It is the most frequently used procedure and, in view of its simplicity, one of the most accurate.

The accuracy of the relative-humidity observation is dependent upon the accuracy of the temperature observations and the accuracy of the tables or charts from which the values are taken. An error of less than 1 per cent of relative humidity can be expected when careful observations are made. This procedure applies between a wet-bulb temperature of 32°F and a dry-bulb temperature of 212°F. Wet-bulb observations can be used to determine air conditions above 212°F, even though relative

humidities are low here. It is basically acceptable below 32°F, but temperatures in this region must be read with exceptional accuracy to secure reliable results.

In cases where the wet bulb may be below 32°F, care must be taken to distinguish between a wet bulb and an ice bulb. As the temperature of the wick drops below 32°F, the water tends to subcool. If a minimum temperature is reached without freezing, wet-bulb tables are valid. However, if freezing occurs, the minimum wet bulb may not be attained, and the temperature rises quickly to 32°F. Observation must be continued until a new minimum is reached, the ice-bulb temperature, for which ice-bulb temperature tables are available.

13.9. Dew-Point Device. The dew-point temperature can be used to establish a state point on the psychrometric chart from

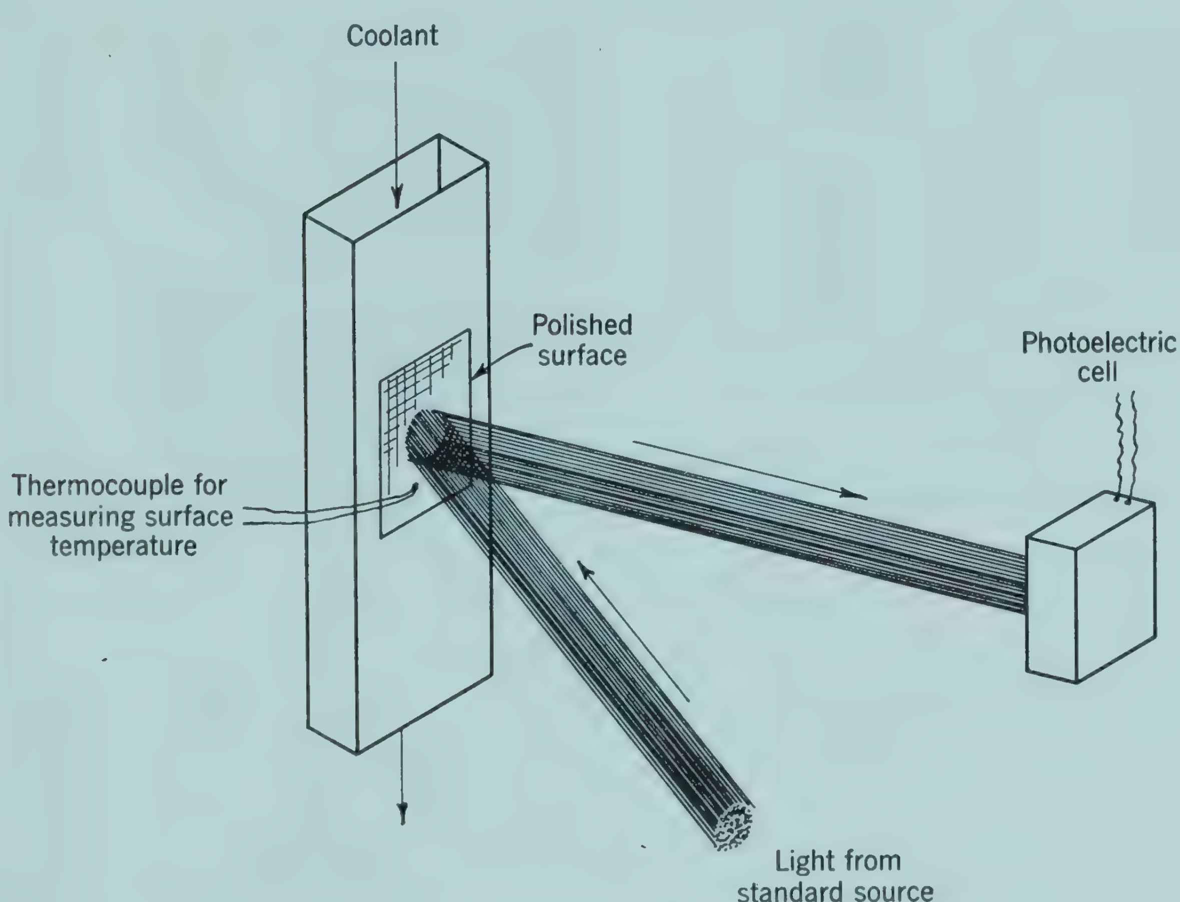


Fig. 13.9. Principle of operation of a dew-point apparatus.

which relative humidity can be taken (sect. 10.12). The temperature of the dew point is observed by lowering the temperature of a polished surface to the point where moisture just starts to condense from the air onto the surface. The exact point or temperature at which this phenomenon starts is difficult to observe

with the naked eye. The procedure outlined in Fig. 13.9 will give more acceptable results.

The light impinging on the photocell is of constant intensity; thus, the photocell signal is constant. The reflecting surface is cooled slowly by the cooling medium. At the instant condensation starts, the light rays are dispersed and the intensity of the signal from the photocell decreases. The surface temperature observed by the thermocouple when the signal decreases in intensity is the dew point.

Accuracies are high. The polished surface must be kept exceptionally clean since dirt will cause the light to disperse, thus confusing the point of condensation. The frost point may be observed to as low a temperature as $-90°F$.

13.10. Hygrometers. Many hygroscopic materials contract and expand significantly when the moisture content varies. Since the moisture content of a hygroscopic material is related to the relative humidity of the surrounding air by the equilibrium moisture curve, the relative humidity of the air can be indicated by the change in dimensions of the material. A device that utilizes such a material is called a hygrometer.

Human hair, wood, and certain animal tissues are the most frequently used materials. The change in length of an element made of these materials is conducted through a kinematic chain to an indicator.

Hygrometers are secondary instruments and must be calibrated against acceptable standards. Temperature, age, and the range of exposure to humidity all affect the calibration. Relative humidities outside the approximate range, 35–90 per cent, are difficult to include in a calibration and, if experienced by a calibrated instrument, may alter the calibration. The instrument's response time is long. Variable humidities may be difficult to follow. Carefully calibrated, used, and handled hygrometers may give satisfactory readings for many observations. However, these instruments must be checked and calibrated frequently if readings of requisite accuracy are to be expected.

13.11. Electric Hygrometer. A hygroscopic salt such as lithium chloride changes its moisture content with relative humidity according to its equilibrium moisture curve. Therefore, an inert material mixed or coated with a salt such as lithium chloride will vary in electrical conductivity with moisture content and

relative humidity. A device using such a material, Fig. 13.10, is available from a number of industrial concerns.

A standard alternating voltage is applied across the unit, and the current noted by an ammeter or by the voltage across a standard shunt for which a potentiometer may be used.

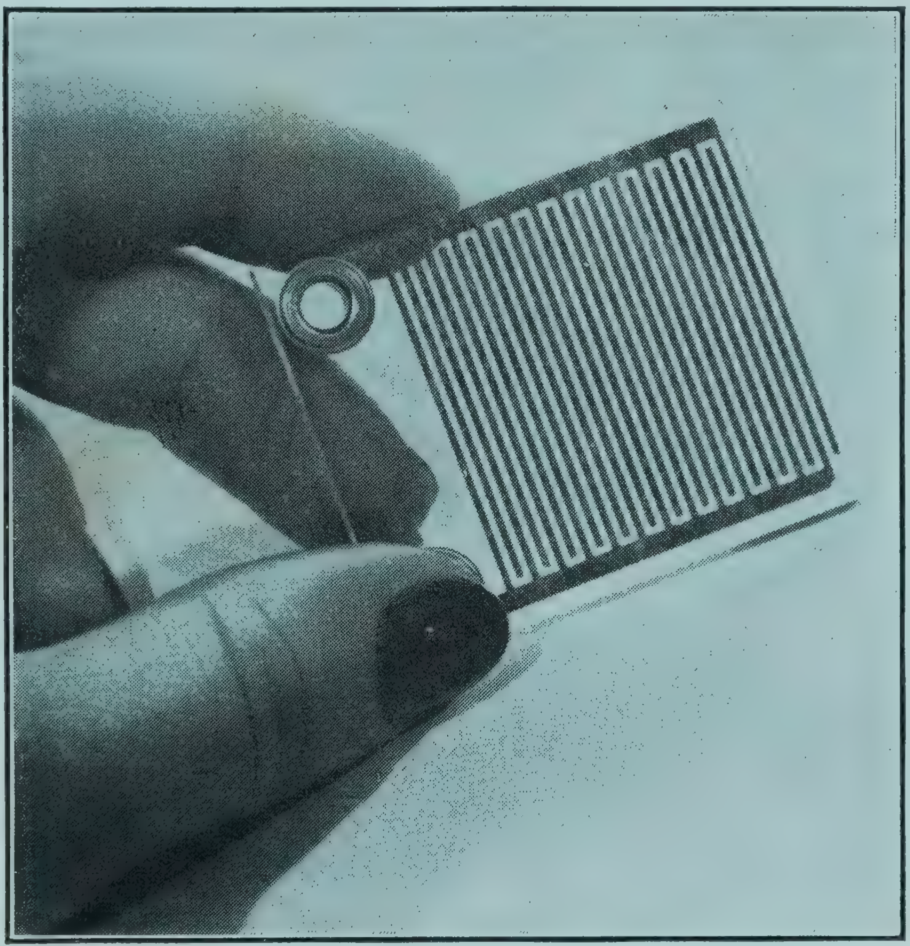


Fig. 13.10. An electric hygrometer sensing element. (*Courtesy* Minneapolis-Honeywell Regulator Co.)

The electric hygrometer is convenient for remote observations and responds quickly. Observations must be corrected for temperature. When properly calibrated and stabilized, this system is accurate to within 1.5 per cent of the reading.

RECORDERS

13.12. Direct Recorders. Temperature and pressure indicators which use bimetal or pressure springs for actuating an indicator needle are available with pen-tipped needles and movable charts upon which a continuous record is made. Pressure spring-bulb temperature recorders are used for recording wet-bulb

temperatures by fitting the bulb with a "sock" that simulates the wet-bulb thermometer. Hair and wood hygrometers are similarly fitted for recording.

13.13. Indirect Recorders. Thermocouple systems involve potentials too small to overcome the friction of a recording system. The galvanometer needle is used as a positioner for a mechanical system that activates a recorder. The millivolt potential may also be fed into a vacuum-tube system (an amplifier) that controls sufficient electrical energy to operate a recording mechanism. The electrical hygrometer has insufficient signal for direct recording and must be combined with a secondary recording system.

13.14. Characteristics. Recorders have approximately the same accuracy as indicators that use a comparable mechanism. Recorders are made with discs, cylinders, and continuous charts. Charts cover various time ranges such as 1 hr, 24 hr, 1 week, 1 month, etc.

CONTROLLERS

13.15. On-Off Controllers. Controllers or control procedures may be divided into two groups, (1) on-off controls and (2) modulating controls. On-off control indicates that the medium being controlled flows full or is completely shut off. A modulating controller varies the rate of flow to match the demand. Flow is continuous. Three types of modulating controls will be discussed; they are: (1) floating, (2) self-operating, (3) pneumatic.

Most room thermostats, controls for refrigerators and hot-water heaters, and switches for automatic water systems are examples of on-off control (see Fig. 13.2). All the devices and mechanisms discussed in this chapter are or can be fitted with various electrical switches or valves that make or interrupt a flow. The bimetal and pressure-spring mechanisms are used extensively for switch activation. Such devices are qualified by the range through which they can be adjusted, the differential range, that is, the difference between the on and off position, the adjustability of the differential, and other important features such as maximum current, rate of flow, and corrosion resistance. Thermocouple systems can be used for control by incorporating a switching mechanism in the indicating or recording system.

Controllers used for liquids or with systems or materials having a large heat capacity follow the response of the medium closely. Air-temperature controls may lag behind the response sufficiently to exceed the control point during each cycle and a performance pattern such as that of Fig. 13.11 will result. Operating response can be treated by the procedures outlined previously if the response pattern can be defined mathematically. The operating range can be minimized by making the *on* portion of the cycle as

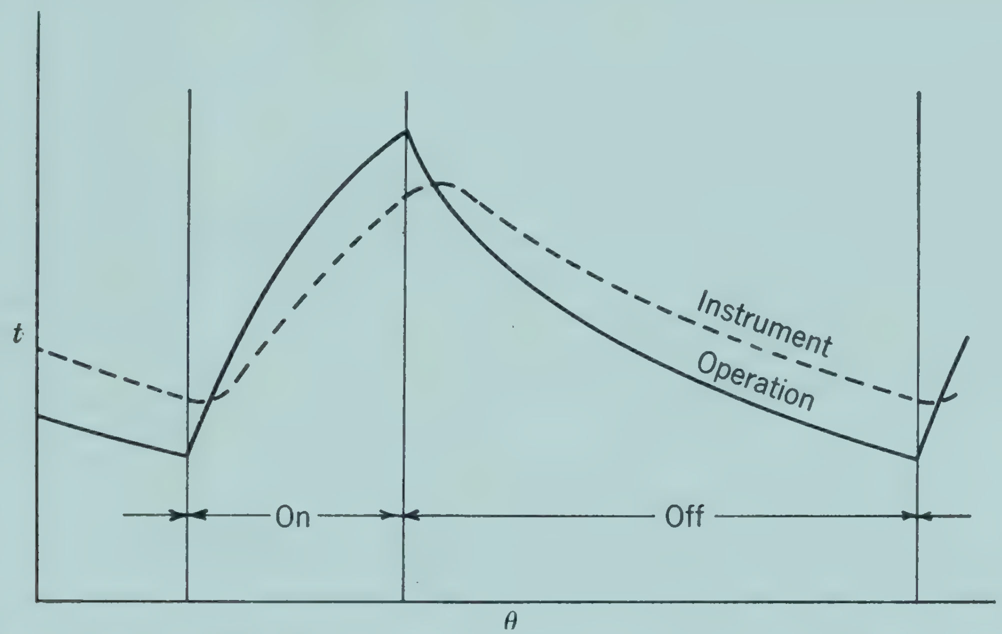


Fig. 13.11. Temperature-time pattern of a recorder following a process controlled by an on-off controller.

long as practicable, by cutting down on the energy input rate, and by using a minimum differential. Further improvement can be provided by dividing the source of heat. One portion is operated continuously, the other by on-off control.

13.16. Floating Controllers. The floating controller is shown in principle in Fig. 13.12. An electric motor is geared to a valve that regulates the flow of a fluid. The two-point switch causes the motor to operate within the control-point differential which positions the valve to handle the load at the prescribed level irrespective of the demand.

As an example, assume that the valve is controlling gas for a burner that is heating a stream of air and that the thermostat is set to operate between 175° and 180°F. When the air temperature drops to 175°F, the closing of the points starts the motor which slowly opens the valve. The valve continues to open slowly until the air temperature reaches 180°F when the other set of

points close, the motor reverses, and the valve closes slowly. Thus the motor slowly oscillates or "floats" the valve within the set range, adjusting the gas rate to the demand.

A motor of this system is also wired to operate only when the points are in contact. Thus, the motor does not operate as long as the temperature is within the set range. When the low point is contacted, the motor operates the valve to bring the system into balance; the motor operates as long as the points are in con-

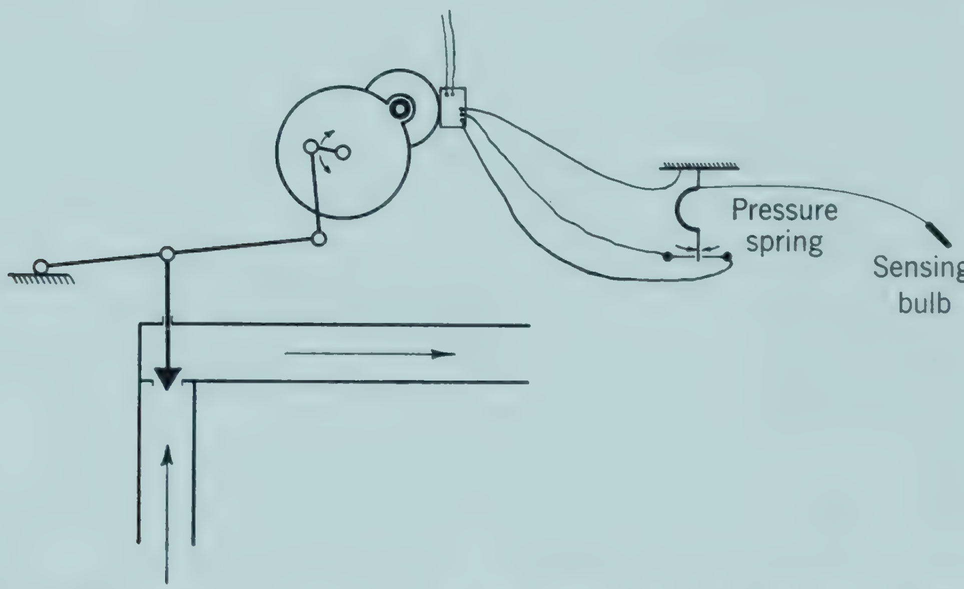


Fig. 13.12. Schematic principle of operation of a floating controller.

tact. The upper contact operates the motor in the reverse direction. This system is used more frequently than the first system discussed.

Care must be used in adjusting the valve-movement rate to the rate of temperature change of the system. A valve-moving mechanism which decreases the rate of valve movement as the set point is approached provides a smoother performance curve.

13.17. Self-Operating Controllers. Controllers of this type (Fig. 13.13) are used extensively because of their simplicity. The bulb is filled with a fluid which has a steep vapor pressure-temperature curve through the operating range so that a maximum valve movement will result for a minimum change in control temperature. The alcohols, ether, and various refrigerants are used to fill the bulb. Valves of this type can be used to control pressure by replacing the pressure bulb with a connection to the pressure being controlled. Note that if the temperature (or pressure) exceeds the control point, the valve will close completely.

A temperature variation of approximately 5°F will operate most valves through their entire range. The response characteristics can be determined by the procedures discussed earlier in the chapter if the valve-shaft friction is not great.

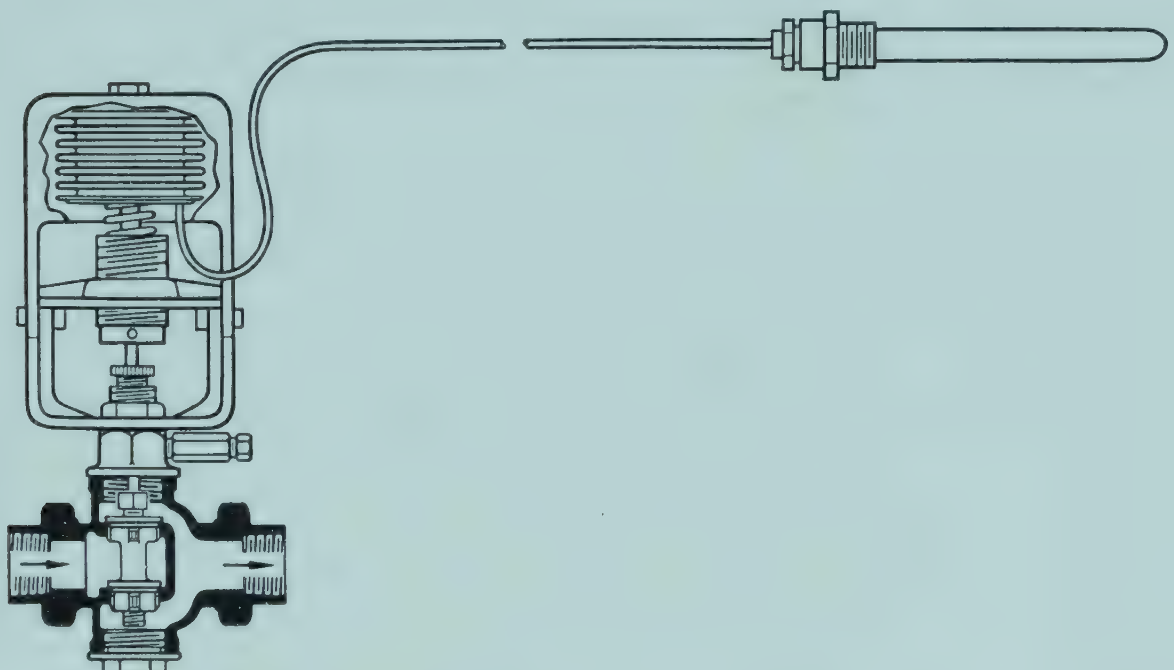


Fig. 13.13. A cross-section drawing of a self-operating controller. (*Courtesy* The Powers Regulator Co.)

13.18. Air-Operated Controls. An air-operated control system is shown schematically in Fig. 13.14. The air system is the most versatile of all the control devices. Since the force or energy required to regulate the control air pressure is small, this system can be combined with any of the indicating devices discussed earlier in the chapter (except the glass thermometer). The sensitivity and rate of response of the instrument can be adjusted through wide ranges by simple adjustments. Thus, it is possible to balance an instrument with the rate of response of a system so that maximum performance will result. The sensitivity can be adjusted to as low as 0.1°F or equivalent in pressure or relative humidity. Since the air motor is basically independent of the control mechanism, air-operated controls can be adapted to a great variety of processes. Availability varies from simple systems with fixed characteristics to systems that are completely adjustable to the characteristics of the process being controlled and that control at a fixed set point irrespective of the magnitude of the load.

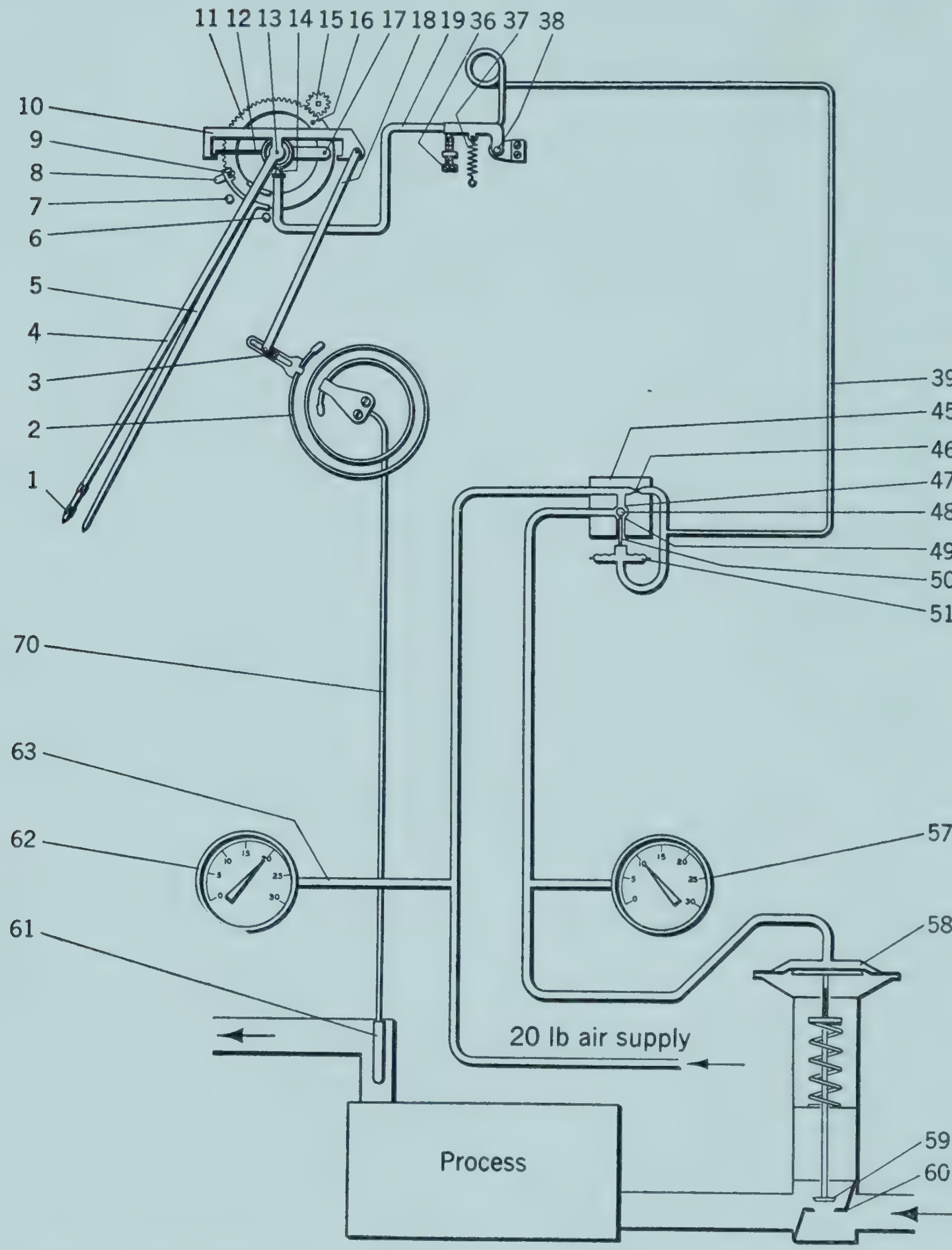


Fig. 13.14. Schematic diagram of a simple air-operated control system. When steady-state conditions exist, air is bled from line 39 through the orifice at 14. The pressure in line 39 positions the ball valve at 48 so that a fixed pressure is produced at 58. Thus, the valve (59) is held at a fixed opening. A change in temperature (61) will cause a change in the orifice clearance (14), and the pressure in line 39 will change. The bellows (51) will adjust the ball valve (48). The pressure at 58 will change, and the valve will move the proper amount to readjust the process condition. (*Courtesy* Taylor Instrument Co.)

High cost is the chief disadvantage of this control. The more sensitive instruments may require expert attention to maintain proper performance.

REFERENCES

1. Behar, M. F. *The Handbook of Measurement and Control.* Instruments Publishing Co. 1951.
2. Dunmore, F. W. An Electric Hygrometer and Its Application to Radio Meteorology. *J. Research Natl. Bur. Standards* (*Research Paper* 1102) 20:723–744. 1938.
3. Eckman, D. P. *Industrial Instrumentation.* Wiley. 1950.
4. Farrington, G. H. *Fundamentals of Automatic Control.* Wiley. 1951.
5. *Fundamentals of Pressure and Temperature Instruments.* Delmar. 1947.
6. *Instruments and Process Control.* Delmar. 1945.
7. Smith, E. S. *Automatic Control Engineering.* McGraw-Hill. 1944.

CHAPTER 14

Cost Analysis

NOMENCLATURE

A = annual replacement reserve or annuity, dollars.
i = interest rate, per cent.
l = expected life, yr.
n = a period of time, yr.
r = annual depreciation rate, per cent.
S = cost new less salvage value, dollars.

The commercial success or failure of a processing enterprise depends, naturally, upon the difference between cost of production and income. In turn, this differential depends upon the integrated performance of the individual units. The over-all economic analysis and management of a plant is a highly skilled process and involves procedures that are too complex to be completely treated here. However, the elementary principles and procedures will be discussed relative to unit operations in order that the engineer will be able to estimate the various costs of simple processes.

The machine of the highest energy efficiency may not be the most satisfactory machine economically. The added air volume from a larger fan may give better performance but not be worth the extra cost. An automatic machine may be expensive but so reduce labor costs that the unit product cost is less. A new sorting device may produce a better-quality, more valuable product, but the increase in unit cost may be greater than the increased value of the products at the existing price structure.

This phase of engineering is too often disregarded. Careful attention should be given to costs since they are one of the most important factors in any engineering problem.

GENERAL CONSIDERATIONS

It is usually advisable to determine the cost analysis on a unit product basis. For example, what is the cost per quart to own

and operate a certain milk-bottle washer? What is the cost per hundredweight to elevate sacked livestock feed with a bag elevator? How much does it cost to quick-freeze 100 lb of meat? Note that the total production cost per unit would be the sum of the unit operation costs.

It is not always advisable for each unit operation to function in the most economical manner. It is the accumulated performance of a series of operations which produces a distinct end point in which we are interested.

Unit costs are closely related to and affected by the "flow" procedure in a plant. This topic will be treated in the next chapter.

ITEMS OF COST

For convenience, the total cost per unit is broken down into (1) fixed and (2) operating costs.

14.1. Fixed Costs are those that are usually not directly related to the amount of use; they include:

1. Depreciation.
2. Interest on the investment.
3. Housing.
4. Taxes and insurance.

The depreciation of many standardized machines for standardized procedures may be a function of use. Steam boilers, choppers, and drag elevators depreciate because of over-all wear. However, since they can be repaired, the length of service can be extended. This is discussed in detail under "life expectancy."

14.2. Operating Costs are those that are directly related to use; they include:

1. Fuel, power, and utilities.
2. Labor.
3. Maintenance.

A machine or unit operation includes the device that produces the result and the power unit that drives the device.

14.3. Depreciation. Depreciation may be defined as the decrease in value of a piece of property during a period of time. The decrease is considered from the standpoint of amounts to be set aside each year in order to recover the cost of the unit at the

end of its useful life. This procedure is frequently called amortization. This decrease, which is usually determined on a yearly basis, is charged to the products produced. This charge is legitimate, since the piece of property, which may be a machine, building, or similar unit, is expended only in order to produce a product.

A distinction should be noted between depreciated value, actual value, resale value, and taxable value. The depreciated value, sometimes called the book value, is the current value as shown by the books of the concern. The actual value is based upon the value which the unit adds to the product. The resale value is the best price that could be received for the unit on the open market. A piece of equipment, a hammer mill, for example, may have cost \$600 new. Its present resale value may be \$470. However, its depreciated or book value may be only \$250, but it may be worth \$650 to the enterprise. It is generally advisable to maintain a depreciated value less than the actual value. Thus, decreased consumer demand or other depressing factors such as forced price reduction do not materially upset the over-all stability of the enterprise. The taxable value is a base value used for calculating taxes and may not be the same as either book or actual values.

14.4. Straight-Line Depreciation. The simplest and most widely used procedure for determining the annual depreciation charge is the straight-line method shown in Fig. 14.1. The depreciation charge per year is

$$\text{(Cost new} - \text{Salvage value)/Total expected life in years} \qquad (14.1)$$

The salvage value is the junk or resale value at the end of the useful life of the machine.

The annual depreciation as determined by the straight-line method if accumulated during the life of the unit is sufficient to replace it at the end of its useful life. This procedure is not too realistic because the accumulated reserve is not credited with interest.

A businesslike, sound procedure which is frequently used is the accumulation of a sinking fund on an annuity basis. A certain amount that is not as large as the annual depreciation based upon the straight-line method is put aside each year. The accumulated amount with interest that compounds annually is sufficient to provide replacement at the end of the useful life. The amount A

that must be placed annually (end of year) at compound interest i for a term of l years equal to the expected life of the unit to create an amount S is:

$$A = S \frac{i}{(1+i)^l - 1} \tag{14.2}$$

S is equal to the cost new less the salvage value. This is a procedure that can be used to provide a sinking fund large enough to recover the cost of the unit by the end of its useful life.

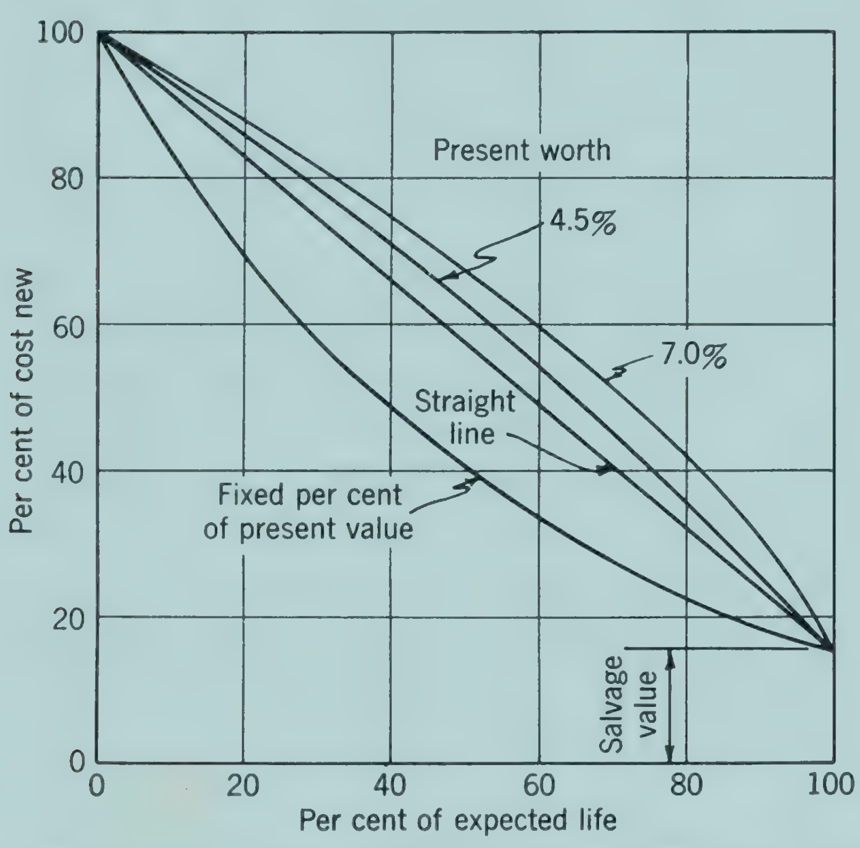


Fig. 14.1. Depreciation of a unit by straight-line, fixed-percentage-of-present-value, and present-worth methods (at two percentage rates).

Consider a \$1000 unit expected to last 10 years (no salvage value). The annual depreciation would be \$100. The annual annuity A would be

$$A = 1000 \frac{0.04}{(1+0.04)^{10} - 1} = \$83.33$$

Thus, only \$83.33 would have to be placed aside annually to accumulate the needed amount in 10 years.

In a going concern containing many units, funds made available by straight-line-depreciation methods are reinvested in the busi-

ness. This not only provides a general depreciation reserve but also permits a satisfactory return on funds that are provided for replacement.

14.5. Other Methods for Calculating Depreciation. Of a number of other methods that are sometimes used for determining the depreciated value, and of course the annual depreciation, two will be discussed.

1. *The Fixed-Percentage-of-Present-Value Method* is sometimes used where it is advisable or desirable to depreciate the unit at a faster rate when new than when it becomes older. Each year, the annual depreciation is a constant percentage of the depreciated value at the beginning of the year. In Fig. 14.1 the constant percentage curve is produced by deducting 16.7 per cent of the value at the beginning of each year as the annual depreciation.

This procedure can be used under economic hazards where it is advisable to depreciate the unit at a fast rate. For example, a dairy-products processer may purchase a homogenizer for fluid market milk. If the demand for homogenized milk is unstable and there is a possibility that conditions may change in the near future, thus reducing the sales of the product, then this depreciation procedure would be advisable.

The constant percentage curve, Fig. 14.1, is based upon the salvage value and is expressed thus:

$$\text{Cost new } (1 - r)^l = \text{Salvage value} \tag{14.3}$$

where r = annual depreciation rate, per cent.
l = expected service life, yr.

After the value of r is determined on the basis of new cost, expected service life, and salvage value, the value at the end of any year n would be

$$\text{Value}_n = \text{Cost new } (1 - r)^n \tag{14.4}$$

Both annual interest on investment and annual depreciation are higher during the first few years of use than later. Consequently, the net operating return during the first years of use must be relatively high to offset these high costs.

2. *The Present-Worth Method* of depreciation is recommended where the factors involved are relatively stable. This would

imply that the cost new would not vary materially over a period of years, that the estimated length of service would be reliable, and that the salvage value could be estimated closely.

The present-worth method is based upon the sinking-fund procedure for providing a replacement reserve, and it may be described as a method of arriving at the present worth or value of expected returns. At an interest rate i, the income for l years from an investment A compounded annually is

$$\text{Income}_l = A[(1 + i)^l - 1] \tag{14.5}$$

This return could be assumed to be from a machine, process, or plant as well as from a like sum of money. The present value of this investment or expended value of the machine is

$$\text{Present value}_n = A[(1 + i)^n - 1] \tag{14.6}$$

Therefore, the present worth or value of expected service or returns is the difference between 14.5 and 14.6. Dividing this by the total expected income (14.5) and multiplying by 100 gives the following expression

$$\text{Condition per cent} = 100 - 100\,\frac{(1 + i)^n - 1}{(1 + i)^l - 1} \tag{14.7}$$

This is known as the condition per cent, which, when multiplied by the cost new, gives the present value or, more exactly, the present worth of expected returns.

Note (Fig. 14.1) the present-worth curves for two rates of interest. These curves yield an annual depreciation which is small at the beginning and increases as the unit gets older. Note that the depreciation for the first year is the amount of the annuity which must be placed annually to provide a replacement amount at the end of the useful life.

The sum of the annual depreciation at any age and interest on the depreciated cost at that age is constant and is equal to

$$\text{Cost new}\left(i + \frac{i}{(1 + i)^l - 1}\right) \tag{14.8}$$

This feature is specially noteworthy and acceptable since this sum by other methods is higher during the earlier years of life.

14.6. Life Expectancy. Depreciation must be based upon an estimated or assumed useful length of service. Suitable values are

frequently difficult to secure. Many studies have been made of the expected useful life or life expectancy of industrial and agricultural equipment. The mortality curves of pieces of industrial and agricultural equipment are similar to human mortality curves used by life insurance companies. This is especially true for those standardized pieces of equipment used under consistent conditions. Railroad rolling stock, power-generating equipment, and meters are examples. Many pieces of agricultural equipment also show this characteristic and the expected life of each determined by systematic studies is generally reliable when the piece of equipment is not affected by unforeseen hazards.

Factors that may cause the actual useful life to be shorter than the estimated or expected life are:

1. Obsolence.
   a. Technological improvement.
   b. Improvement in machine design.
2. Change in supply or consumer demand.
3. Change in production factors, such as:
   a. Labor.
   b. Power.
   c. Transportation.

Table 14.1 lists some expected-life values for a selected number of units of equipment. The amount of use may affect the expected life of a unit. However, since systematic maintenance and repairs can prolong the usefulness of a unit, an exact life expectancy is difficult to provide. The manager must balance annual repair and maintenance costs against the added depreciation of a new unit. Since costs vary, each case must be handled separately.

Considerable judgment must be exercised in selecting an expectancy value for a specific case. The best procedure is to inspect the unit after a period of years, assigning new and more reliable values. A straight-line depreciation curve which was reevaluated is shown in Fig. 14.2. Note that the expected life and salvage value have both been changed. This is a recommended procedure since current conditions as indicated above may change the length of the useful life as estimated initially.

14.7. Interest on Investment. Interest on investment is charged against the piece of equipment as an initial or guaranteed return to owners and investors, such as bond, stock, or mortgage

Table 14.1 EXPECTED SERVICE LIFE OF SOME UNITS USED IN PROCESSING

Unit	*Expected Service Life, yr*	*Source*
Feed grinder	19	Davidson and Henderson [4]
Frame building	20	Roadhouse and Henderson [6]
Brick building	33	Roadhouse and Henderson [6]
Boiler and settings	12	Roadhouse and Henderson [6]
Pumps	7–10	Roadhouse and Henderson [6]
Oil burners	7	Roadhouse and Henderson [6]
Water softeners	8	Roadhouse and Henderson [6]
Refrigeration machines	13	Roadhouse and Henderson [6]
Air compressors	20–25	J. H. Perry [5]
Conveyors, belt	15	J. H. Perry [5]
Conveyors, chain	20	J. H. Perry [5]
Conveyors, screw	10	J. H. Perry [5]
Crushers	12–17	J. H. Perry [5]
Driers	20	J. H. Perry [5]
Elevators, bucket	20	J. H. Perry [5]
Fans	10–20	J. H. Perry [5]
Furnaces	20	J. H. Perry [5]
Separators, centrifugal	18	J. H. Perry [5]
Motors, a-c	20	J. H. Perry [5]
Packing machinery	17–20	J. H. Perry [5]

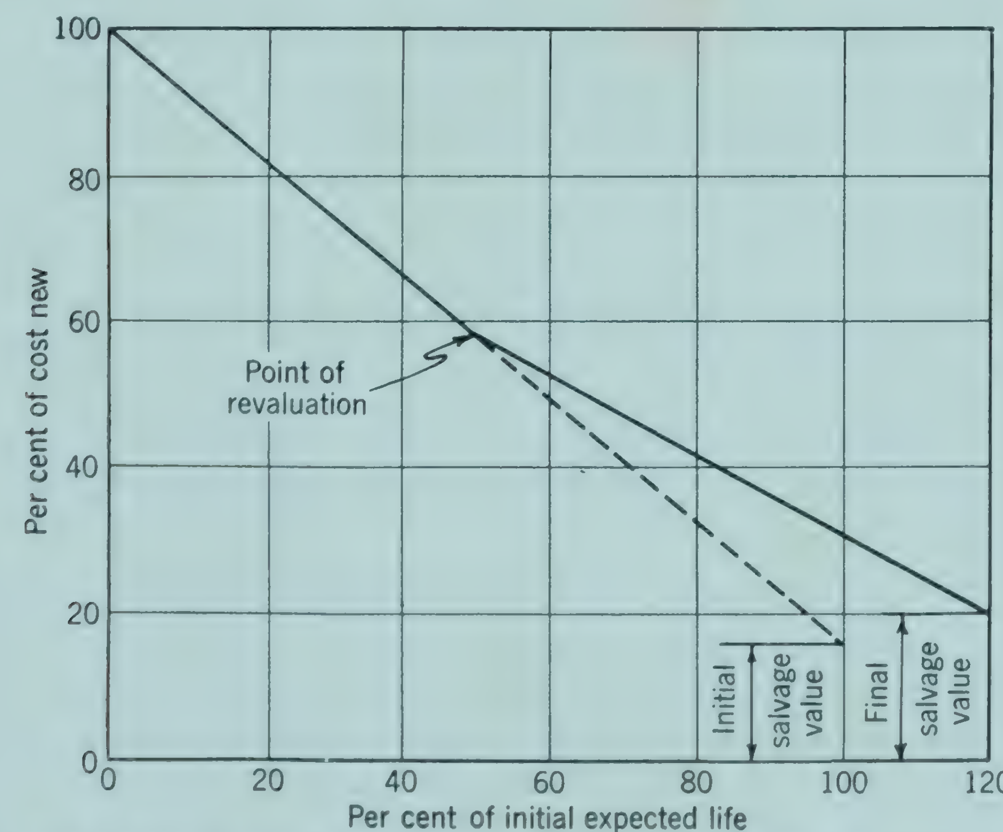


Fig. 14.2. Straight-line depreciation showing the result of reevaluation.

holders with a contract rate of interest on the investment. When management is the owner and the piece of equipment is not mortgaged or otherwise encumbered, interest is still charged to the unit as minimum or guaranteed return to the owner. If a unit cannot return more than would be received if the money invested were loaned at the current rate of interest, then its use is questionable.

Interest on investment is determined in three different ways. Consider a unit costing \$1000 and expected to last 10 yr with the current rate of interest being 4 per cent.

1. *Interest on Depreciated Value.* The depreciated value at the beginning of the year is used as the principal. For example, during the third year the above unit would be worth \$800 and the interest would be \$32. Similarly, the value during the eighth year would be \$400 and the interest, \$16. This procedure is recommended when the unit is of major importance or otherwise is separate and distinct from other units and retains its identity during its entire life. A steam boiler, diesel engine, or large hammer-mill would fall in this class. There might be only one of each in any plant, and it would be replaced at the end of its useful life.

2. *Interest on Half of Cost New (Used with Straight-Line Depreciation).* In this way the principal used each year is half the cost or \$500, referring to the above example, and the annual interest charge would be \$20. This is the most frequently used method and the most practical for most plants. It is particularly adaptable for enterprises where the identity of each unit does not need to be maintained and the interest charged eventually reverts to the enterprise itself. A further distinctive feature is that all the pieces of equipment do not need to be replaced at the same time. The enterprise as such may be mortgaged, but the indebtedness applies to the enterprise as a whole rather than to individual units or machines.

3. *Interest on Total Cost New.* Infrequently, the annual interest charge is based upon the cost new. This may be justified in cases where the expected life cannot be well defined or peculiarly hazardous conditions exist. It is not recommended for general use.

14.8. Housing. It may or may not be necessary or advisable to include a fixed annual charge for housing. It is usually customary to handle housing as a separate and distinct item entirely separate from the production units. In spite of this procedure, housing is a direct result of the necessity for protecting the equipment. Consequently, it should be prorated among the various duties that it performs which would be storage, office and shop space, space for equipment, supplies, miscellaneous.

The items of cost of the housing unit can be broken down in the same manner as for a machine, calculated on a yearly basis, and then prorated on the basis of floor area, structural volume occupied, value added to the product, or cost of the unit. It is usually advisable to base the calculations on the floor area. However, if the plant is organized on a tier basis with units located above each other and as close together as possible, volume occupied may be a more accurate dividing index than floor area. In certain instances where a small but expensive machine adds much to the value of the product, housing charges may justifiably be divided on the basis of relative value added to the product, annual value of the unit, or cost of the unit. For example, a sorting table costing $600 used in a peach-processing plant may utilize an area of 120 sq ft. An automatic pitting machine which costs $3500 may occupy only 20 sq ft. It would probably be more equitable to prorate housing on a cost-new basis rather than on a floor-area basis in this instance because of relative worths of the two units.

Note sect. 14.14 for an example of the determination of these charges.

14.9. Taxes and Insurances. Taxes are based upon the assessed value and are determined by a certain number of mills on each assessed dollar of valuation (a mill is one-tenth of a cent). Insurance is based upon the current value, and the rates are on the basis of each $100 value.

14.10. Fuel, Power, and Utilities. As here used, power includes the fuel used for a prime mover or boiler. Utilities would include water, compressed air, etc. In some cases it would be difficult to differentiate between these, but this presents no problem since charges are determined in the same manner in each case. Electricity, steam, gas, fuel oil, gasoline, compressed air,

and hot and cold water would be included if supplied from a source outside the unit being considered.

Charges are most conveniently made on a daily or hourly basis. For example, electricity may be determined on a kilowatt-hour per hour basis. The amount of water for a washer may be determined per day, or perhaps per hour if the rate of flow is known. Where certain utilities, steam for example, produced in a plant are used in more than one operation, charges should be prorated on a quantity basis.

14.11. Labor. Labor charges apply to operations and can usually be made on a daily or hourly basis. Labor should be carefully allocated to the machine. If a supervisor is tending a number of semiautomatic machines, his cost should be allocated on the basis of the actual time given to each machine. On the other hand, if a machine requires a man (or men) to be on duty even at times when not operating, the total charges must be applied to the machine.

14.12. Maintenance. The following would be included under maintenance:

Lubrication.

Normal replacements due to wear.

Repairs resulting from unforeseen accidents or unexpected failures.

Skilled labor for special services such as adjustments or special repairs.

Painting or cleaning.

Repairs, labor, and painting and cleaning can usually be determined on a yearly or other fiscal basis. Lubrication and normal replacements are related more directly to use so they can be estimated on the basis of production. These charges should be estimated as accurately as possible. Frequently they are valued as a percentage of the cost new per year, 1 to 5 per cent being the range usually experienced.

14.13. Illustration of Analysis. (*a*). A farmer who grinds about 500 bushels of grain annually is considering purchasing an 8-in. burr mill that will grind 55 bu per hour and will require a 6-hp motor. The cost is $150, no salvage, probable life, 19 yr. Maintenance is expected to cost $0.50 per 100 bu. What would be his grinding cost per bushel?

Fixed Costs

Depreciation = Cost new/Probable life	
= \$450/19 =	\$23.60 per yr
Annual interest charge = Half of cost new × Rate	
= \$450/2 × 4% =	9.00 per yr
Housing (nil)	
Taxes and insurance (Taxes are 20 mills per dollar of assessed value which is 60% of actual value. Insurance is about 30¢ per \$100 per year. Therefore, taxes and insurance would amount to 1.5%.)	
= Half of cost new × Combined rate	
= \$150/2 × 1.5% =	3.36 per yr
Total annual fixed cost	\$35.96
Fixed cost per bushel	7.2¢

Operating Costs

(Consider operating charges on an hour basis * during which time 55 bu will be ground)	
Power = 6 hp-hr, 6 kwhr at 2¢ * =	0.12
Labor = 1 man-hr at \$1.25 =	1.25
Maintenance = \$0.50 × $\frac{55}{100}$ =	0.27
Total hourly operating cost	\$2.46
Operating cost per bushel	4.8¢
Total cost per bushel	12.0¢

* For power and power cost estimates, 1 kwhr per hp may be assumed. This assumption presupposes a motor efficiency of 74.6 per cent.

14.14. Illustration of Analysis. (*b*). A small producer-distributor dairy enterprise must change from raw milk to pasteurized milk. In order to do this, the operator must purchase a pasteurizer, a sanitary pump for emptying the pasteurizer, a steam boiler to heat the pasteurizer, and a second and larger surface cooler to cool the milk as it comes from the pasteurizer. How much will this operation add to the cost of production per quart?

The vital data concerning the pasteurizer are:
Maximum volume, 90 gal per day.
Average volume, 70 gal per day, including 15 gal cream.
Operating time per day, 1 hr.

Pasteurizer data:
100 gal, vertical, circular, jacket heated, with propeller agitator; installed cost, \$2000; salvage value, 5% of initial cost; probable life, 8 yr; maintenance, 3% per yr; floor space, 6 x 8 ft; 10-hp boiler required; ¼-hp motor, averaging ⅔ load.

Pump data:
1 in.; ¼ hp; installed cost, \$140; salvage value; 5% of initial cost; probable life, 8 yr; maintenance, 3% per yr, set up in pasteurizer floor area.

Boiler data:

10 hp; installed cost, \$400; oil burner, \$150; automatic controls, \$150; salvage value, 6%; probable life, 12 yr; maintenance, 2½% per yr; floor space, 4 x 5 ft. This boiler will replace a hot-water boiler which is used for heating wash and utility water. Steam will be used for heating the office building and for washing operations. Consequently, its cost must be prorated among the operations it serves.

Surface cooler data:

1500 lb milk per hr; cost, \$500; probable life, 13 yr; salvage value, 2½%; maintenance, 2%; 4500 lb tap water per hr; 6000 lb chilled water per hr; space occupied, 3 x 8 ft; refrigeration charge, 50,000 Btu (0.17 ton days or 8 kwhr). These are the amounts required to cool the milk as it is pumped from the pasteurizer which takes 50 min.

Cost Analysis:

Pasteurizer

$$\text{Depreciation} = \frac{\text{Cost new} - \text{Estimated salvage value}}{\text{Probable life}}$$

$$= \frac{\$2000 - 5\% \text{ of } \$2000}{8} = \qquad \$237.50 \text{ per yr}$$

Interest = One half (Depreciable cost + Estimated salvage value) × Interest rate

$= [\frac{1}{2}(\$2000 - 5\% \text{ of } \$2000) + 5\% \text{ of } \$2000]4\% =$ 41.00 per yr

Housing, 48 sq ft at \$1.25 = 60.00 per yr

Taxes and insurance = One half (Depreciable cost + Estimated salvage value) × Combined rate

$= [\frac{1}{2}(\$2000 - 5\% \text{ of } \$2000) + 5\% \text{ of } \$2000] \times 1.5\% =$ 15.75 per yr

Total annual fixed cost	\$354.25
Fixed cost per quart	0.35¢

Power and utilities = $\frac{1}{4}$ hp Motor for 1 hr per day. $\frac{1}{4}$ kwhr at 2¢. (Note that steam costs are accounted for in the boiler analysis) = \$ 0.01 per day

Labor = 1 hr per day at \$1.00 = 1.00 per day

$$\text{Maintenance} = \frac{3\% \times \$2000}{365} = \qquad 0.17 \text{ per day}$$

Total daily operating cost	\$1.18
Operating cost per quart	0.42¢

Pump

$$\text{Depreciation} = \frac{\$140 - 5\% \text{ of } \$140}{8} = \qquad \$17.10 \text{ per yr}$$

Interest = $[\frac{1}{2}(\$140 - 5\% \text{ of } \$140) + 5\% \text{ of } \$140]4\% =$ 2.94 per yr

Housing (none claimed) ——

Taxes and insurance = $[\frac{1}{2}(\$140 - 5\% \text{ of } \$140) + 5\% \text{ of } \$140] \times 1.5\% =$ 1.10

Total annual fixed cost	\$21.04
Fixed cost per quart	0.02¢

Power and utilities = $\frac{1}{4}$-hp motor for 1 hr at 2¢ = 0.01 per day

Labor = $\frac{1}{2}$ hr at \$1.00 = 0.50 per day

$$\text{Maintenance} = \frac{3\% \times \$140}{365} = \qquad 0.01 \text{ per day}$$

Total daily operating cost	\$0.52
Operating cost per quart	0.19¢

Boiler

Depreciation $= \dfrac{\$700 - 6\% \text{ of } \$700}{12} =$	54.80 per yr
Interest $= [\frac{1}{2}(\$700 - 6\% \text{ of } \$700) + 6\% \text{ of } \$700]4\% =$	14.84 per yr
Housing = 20 sq ft at \$1.25 =	25.00 per yr
Taxes and insurance $= [\frac{1}{2}(\$700 - 6\% \text{ of } \$700) + 6\% \text{ of } \$700] \times 1.5\% =$	5.56 per yr
Total annual fixed cost	\$100.20
Amount charged to pasteurizer *	33.40
Fixed cost per quart	0.03¢
Power and utilities = 2 gal fuel oil at 8¢ =	0.16 per day
Electricity for feed water pump and oil burner $1\frac{1}{4}$ kwhr at 2¢ =	0.03 per day
Labor (none)	
Maintenance $= \dfrac{2\frac{1}{2}\% \times \$700 \times \frac{1}{3}}{365} =$	0.02 per day
Total daily operating cost	\$0.21
Operating cost per quart	0.07¢

Surface Cooler

Depreciation $= \dfrac{\$500 - 2\frac{1}{2}\% \text{ of } \$500}{13} =$	\$37.45 per yr
Interest $= [\frac{1}{2}(\$500 - 2\frac{1}{2}\% \text{ of } \$500) + 4\frac{1}{2}\% \text{ of } \$500]4\% =$	10.25 per yr
Housing = 24 sq ft at \$1.25 =	30.00 per yr
Taxes and insurance $= [\frac{1}{2}(\$500 - 2\frac{1}{2}\% \text{ of } \$500) + 2\frac{1}{2}\% \text{ of } \$500]1.5\% =$	3.84 per yr
Total annual fixed cost	\$81.54
Fixed cost per quart	0.01¢
Power and utilities = Water, 540 gal at 50¢/1000 =	0.27 per day
Refrigeration † 8 kwhr at 2¢ =	0.16 per day
Labor $= \frac{1}{2}$ hr at \$1.00 =	0.50 per day
Maintenance $= \dfrac{2.0\% \times \$500}{365} =$	0.03 per day
Total daily operating cost	\$0.96
Operating cost per quart	0.34¢

Summary of costs per quart

Pasteurizer	Fixed cost	0.35¢ per qt
	Operating cost	0.42¢
Pump	Fixed cost	0.02¢
	Operating cost	0.19¢
Boiler	Fixed cost	0.03¢
	Operating cost	0.07¢
Cooler	Fixed cost	0.01¢
	Operating cost	0.34¢
	Total	1.43¢

* It is estimated that ⅔ of the steam produced by the boiler will be used for cleaning, washing, and heating and only ⅓ for the pasteurizer. Note that this division will not affect operating charges.

† This is the operating charge. Normally, an allocate part of the fixed costs would be charged to the cooler, but this was neglected because the cooler is a very small part of the total load handled by the refrigerator.

These examples demonstrate a very important feature of costs; the more a piece of equipment is used, up to a certain point, the less it costs per unit produced. We assume that fixed costs are

not a function of the amount of use. This is essentially true except that a piece of equipment may have a shorter life if used extensively. On the other hand, actual life is affected by many factors as previously indicated so that the actual weight of use in this consideration is small. Therefore, the use, unit cost, annual fixed-cost relationship would be nearly

$$\text{Fixed cost per unit} = \frac{\text{Total annual fixed cost}}{\text{Units produced per year}} \tag{14.9}$$

It is true that a machine that is used extensively generally requires greater maintenance expense per unit as it ages than one

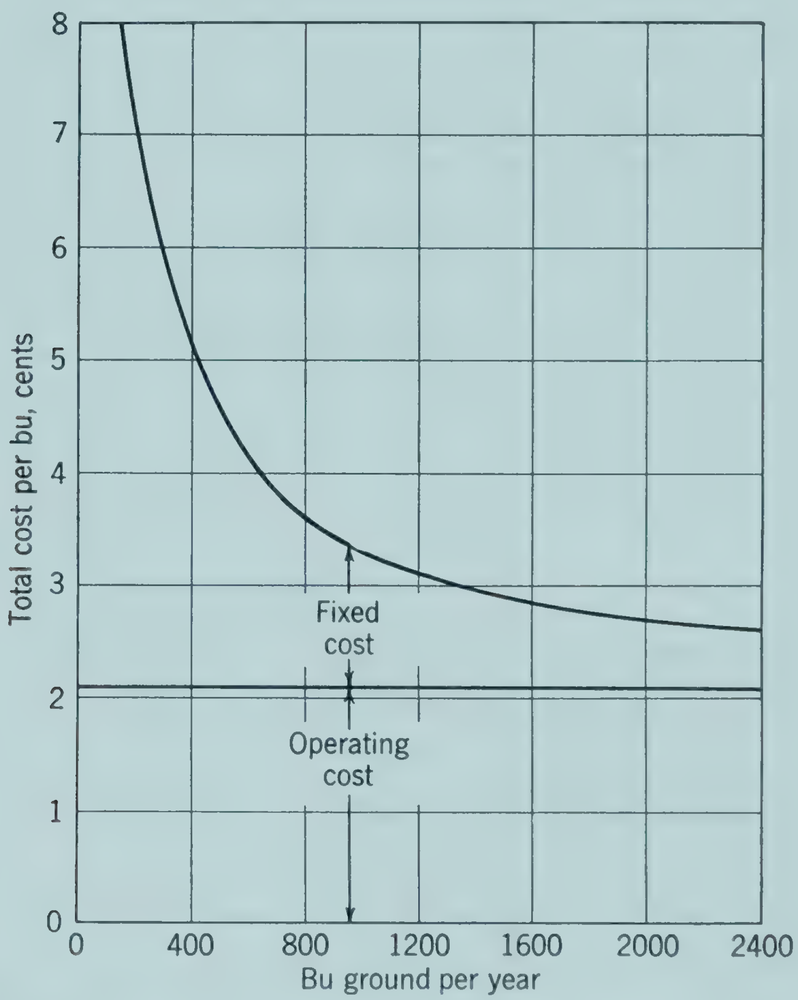


Fig. 14.3. Relationship of total cost per unit to amount of use per year.

used less. However, power and labor costs are usually much greater per unit so that any increase in maintenance cost produces a small if significant increase in the operating cost per unit. Constant operating costs can be assumed without seriously affecting the validity of the results.

This relationship for the feed grinder is shown in Fig. 14.3. Irrespective of the amounts of use the operating cost is constant, whereas the fixed cost is hyperbolic.

These examples show that labor is a big cost item. This is true in general. Better use of labor is usually the most effective way for lowering operating costs.

14.15. Selection, General Principles. The selection of a new piece of equipment or the decision to replace an old piece of equipment cannot always be based upon a definite procedure. Frequently, judgment must be relied upon heavily. The procedure outlined in this chapter can be used for close estimation of unit costs, but judgment must be used when such factors as future prices, consumer demand, and raw-materials supply must be considered.

REFERENCES

1. Armstrong, C. V. Industrial Property Records for Accounting and Valuation Uses. *Ia. Eng. Expt. Sta. Bull.* 160. 1944.
2. Bullinger, C. E. *Engineering Economic Analysis.* McGraw-Hill. 1942.
3. Davidson, J. B. Life, Service and Cost of Service of Farm Machinery. *Ia. Agr. Expt. Sta. Bull.* 260. 1929.
4. Davidson, J. B., and Henderson, S. Milton. Life, Service and Cost of Service of Farm Machines on 400 Iowa Farms. *Ia. Agr. Expt. Sta. Bull.* P. 37. 1942.
5. Perry, J. H. *Chemical Engineers Handbook.* Third Edition. McGraw-Hill. 1950.
6. Roadhouse, C. L., and Henderson, J. L. *The Market Milk Industry.* First Edition. McGraw-Hill. 1941.
7. Winfrey, Robley. Depreciation of Group Properties. *Ia. Eng. Expt. Sta. Bull.* 155. 1942.
8. Winfrey, Robley. Statistical Analysis of Industrial Property Retirements. *Ia. Eng. Expt. Sta. Bull.* 125. 1935.
9. U. S. Treasury Department. Income Tax Depreciation and Obsolescence. Estimated Useful Lives and Depreciation Rates. *U. S. Bur. Internal Revenue Bull.* F. U. S. Government Printing Office.

CHAPTER 15

Process Analysis and Plant Design

The performance of a processing plant depends upon the efficiency with which the material flows through the plant. Efficiency as here used refers to the accumulated efficiencies in the use of individual machines, labor, power, utilities, storage and work space, roads and transportation, and other facilities necessary for production.

The various activities that constitute a process must be so integrated that the movement of material through the plant proceeds smoothly and with a minimum of interruption. If the operations are more or less automatic and control personnel only is required, the location of units, conveying systems, and storage areas can be arranged with efficient performance of the units being the prime objective. If the activities are mostly manual, the operations must be set up in light of convenience, safety, and productivity of the workers in addition to efficient machine performance. A process must be so constituted that normal contingencies such as breakdown, irrregular rate of influx of raw products, and change in characteristics of raw products will not disrupt the normal operations. These factors and others of a similar nature must be considered when analyzing a processing operation.

A complete processing plant consists of a number of unit operations or processes arranged in a certain sequence which may be simple, as in feed grinding on the farm, or complex, as in producing cotton-seed oil or soybean oil, for example. In designing a new plant or rearranging an existing plant for more efficient operation, certain established procedures should be followed in order to secure the most satisfactory results. Process charts and

flow diagrams * which will be discussed in this chapter are excellent tools to use for studying an existing processing plant when improvements are needed or for designing a new plant.

PROCESS CHARTS

A process chart is a schematic presentation of a process, showing the events which constitute the process, their order, and certain desired information concerning each. Chart forms are varied since each individual problem is unique. The experienced process analyst is able to tailor the chart to the situation at hand, but the novice should follow certain procedures that are known to be satisfactory in most studies.

There are three types of process charts, although it is not always easy to distinguish between them when in use.

15.1. An Operation Process Chart is a graphic presentation of all the events in a process and their sequence. The time required for each event or operation and the distance between operations may be included if pertinent. Other information may be added if it contributes to the study. The operation process chart is used for studying the entire operation.

15.2. A Flow Process Chart is a graphic presentation of all the operations and all intermediate events showing transportations, inspections, storages, and delays. All information needed to meet the analysis objectives is included, for example, labor required, times, distances, capacities, utilities, temperatures, and other information. The flow process chart may represent the entire process, but because of the volume of detail it is generally used for significant parts of it only.

15.3. The Layout Sheet or Layout Diagram is a graphical plot of operation or event locations showing the direction of flow of materials and labor. It is usually a plan layout to scale showing space utilization and direction of movement of materials. However, elevation and three-dimensional presentations are frequently used.

* Manual operation economy discussed in Chap. 16, is closely related to process efficiency and should be considered when designing a new plant or analyzing a going concern.

15.4. Nomenclature. Studies have shown that a process is composed of five types of steps in various combinations. These steps and symbols which have been standardized by the American Society of Mechanical Engineers [1] follow.

1. *Operation* ○. Any activity that alters the physical or chemical characteristics of a material or an object, or adds to it in any way whatsoever, such as grinding grain, drying hay, weighing eggs, freezing meat, nailing box, mailing letter.

2. *Transportation* ⇒. Any movement of material from one place to another unless such movement is an integral part of an operation, such as milk by pump, refuse by truck, fruit by belt, box by employee, grain by truck.

3. *Inspection* □. An examination by an individual to determine quality or quantity or to verify conditions, such as determining moisture content of grain, determining grade of fruit, checking performance of vegetable washer, noting temperature of pasteurizer.

4. *Storage* ▽. A desirable interruption of activity, such as ingredients being held for future use, fruit being held for optimum market.

5. *Delay* D. An undesirable interruption of activity, such as prunes in tray waiting to be dried, material in truck waiting to be unloaded, employee waiting for machine to operate.

6. *Combined Operation.* Where two or more operations take place simultaneously, the requisite symbols are combined. For example, cheese in a processing room is both a storage and an operation; a mixing process may be both an operation and a transportation.

Process charts may be built around (1) materials, (2) men, or (3) machines. They may show the movement of materials and all operations thereon. They may show the activities of a man or men required to produce a certain end point. Or the activities of a machine or series of machines may be represented.

15.5. Operation Process Charts. An operation process chart of seed preparation is shown in Fig. 15.1. It will be used to demonstrate the methods of chart design and its use.

Operation process charts can be prepared on plain sheets of paper of sufficient size to include all the required operations, or a form such as shown can be used. The chart type is *material, man,*

or *machine,* indicating the type of procedure being charted. The method is *present, proposed,* or *revised,* indicating existing condi-

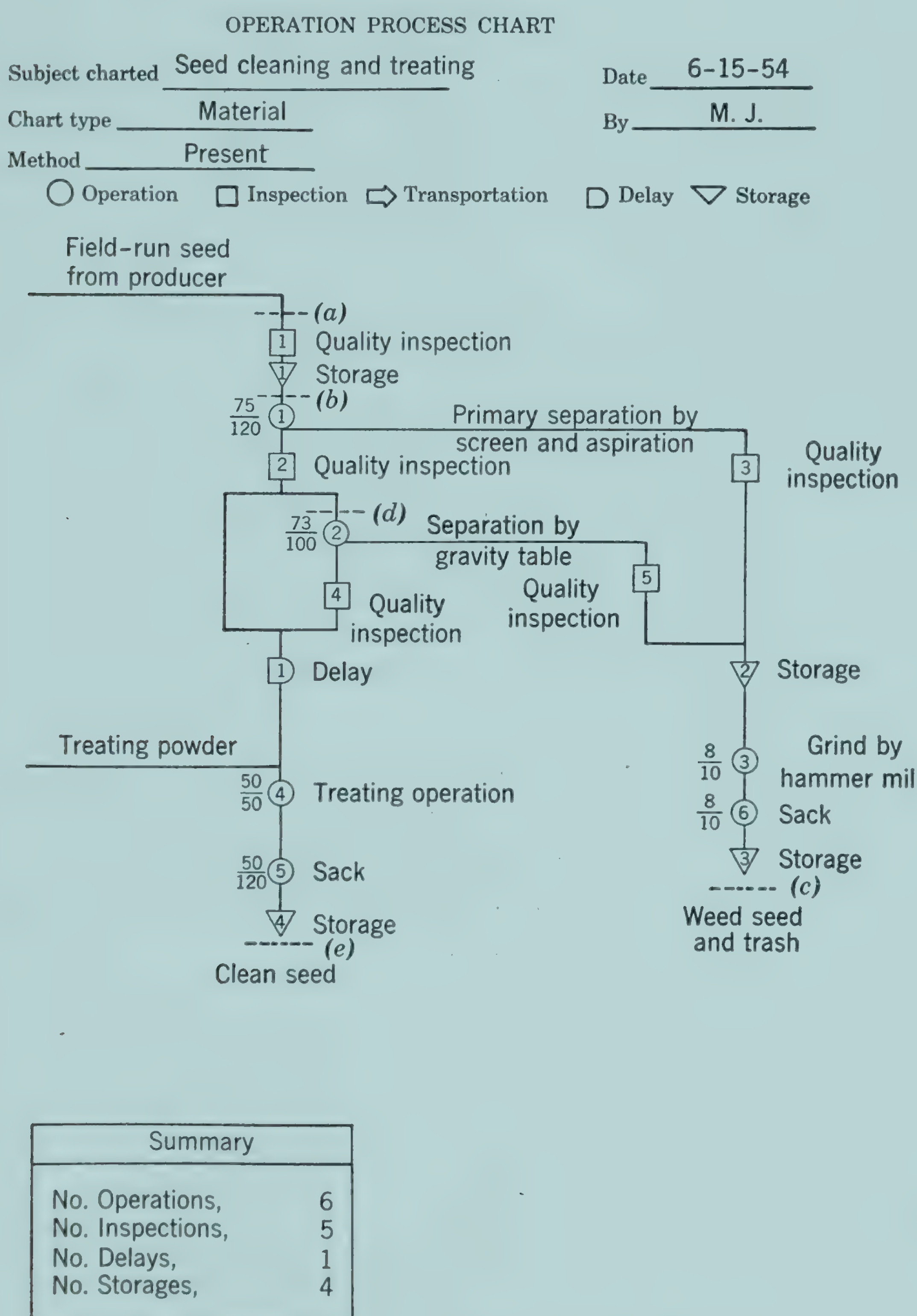


Fig. 15.1. An operation process chart.

tions, proposed changes, or revised procedure developed after previous study. Movement of materials is from left to right on horizontal lines and down on vertical lines unless otherwise indi-

cated by arrows. The main flow should begin at the top of the sheet and proceed straight downward. Material added to the main line of flow is indicated by a horizontal line to the *left* of the main-line flow. Material removed is indicated by a horizontal line to the *right* of the main flow line. Conventional details and their meaning are shown in Fig. 15.2.

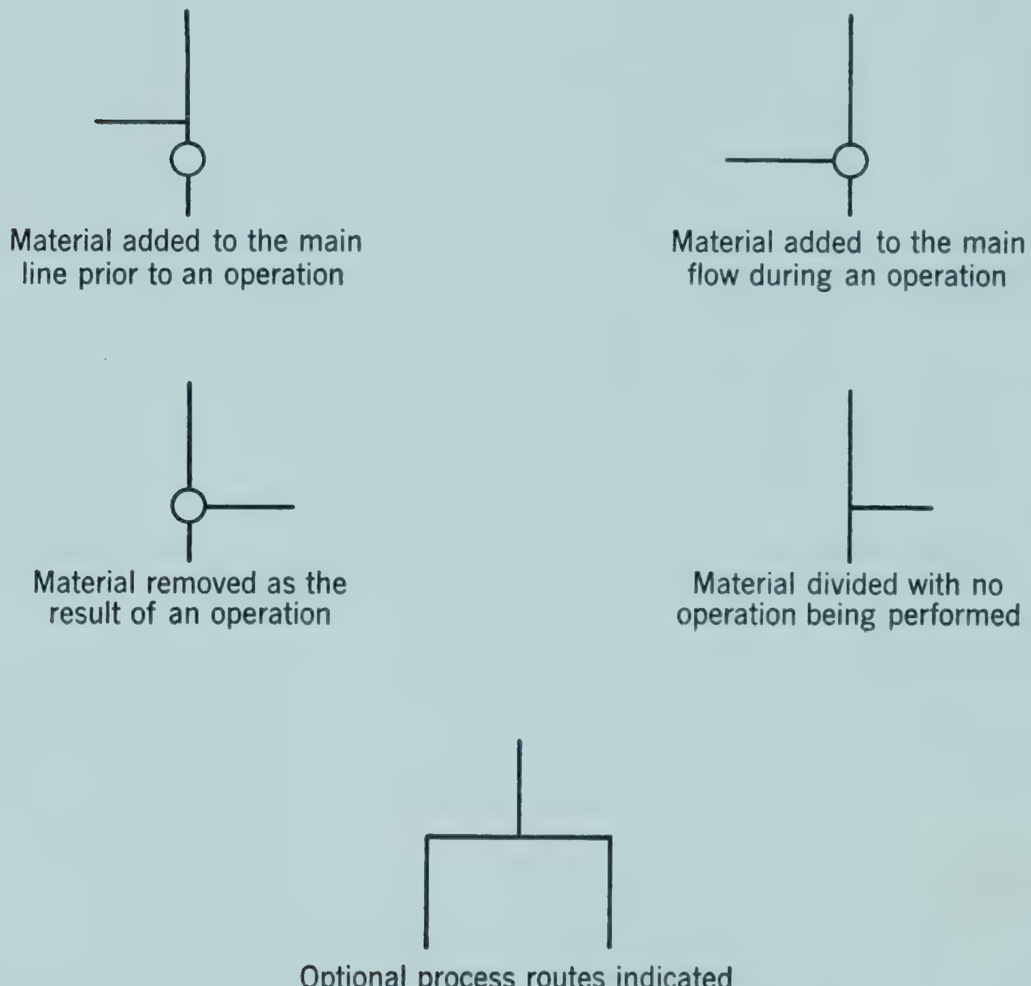


Fig. 15.2. Conventional construction details used in preparing an operation process chart.

The fractions accompanying each operation indicate the actual rate in bushels per hour over the capacity. This is an excellent index for continuous-flow processes. If the material being processed can be segregated into units, a single bird in a poultry-processing plant, for example, the time expended on each unit during each operation is usually used. Time is used only if it contributes to the analysis.

The summary of steps may be omitted, but it shows the balance between the various types of events and usually is an aid to analyzing a process, especially if it is complicated.

The following should be considered relative to an operation process chart.

1. *What Events Can Be Combined?* Combining steps usually reduces labor and may reduce other costs. For example, the machine used on operation two might be substituted for the machine of operation one. This would eliminate inspections four and five and operation two. There would be a saving in labor and amortization of the machine of operation one.

2. *Can Inspections Be Eliminated?* Inspections frequently indicate lack of refinement of an operation or series of operations which could be eliminated by improving operation design or techniques. However, if the raw product is variable it may be inadvisable to attempt to eliminate inspections. Furthermore, it may be economically advisable to use inspections in connection with simple economical operations rather than more expensive procedures that do not require inspections.

In our example, inspection one is made to determine the quality of seed received from the producer in order to determine the price to be paid. The other inspections are necessary to adjust the operation of sorters.

3. *Can Delays Be Eliminated or Converted into Storages?* Delays and storages are similar except a delay is undesirable whereas a storage is desirable. A delay makes it impossible to use men or machines in the most efficient way. A storage may facilitate the use of men and materials.

For example, storage number two retains all the reject from operations one and two. When seed is not being delivered from the producer at the maximum rates, labor can be released from the major operations to grind, sack, and store the reject. The delay is necessary because the treating equipment does not have sufficient capacity. The separating and sacking capacity is 100 bu per hr, the rate being controlled by operation number two. The slower treating operation actually reduces the potential capacity by half.

An operation process chart is usually prepared as a basis for a more detailed study of parts of the chart which are known to need improving. This need may have been recognized before the operation process chart was prepared or may have been discovered through study of the chart.

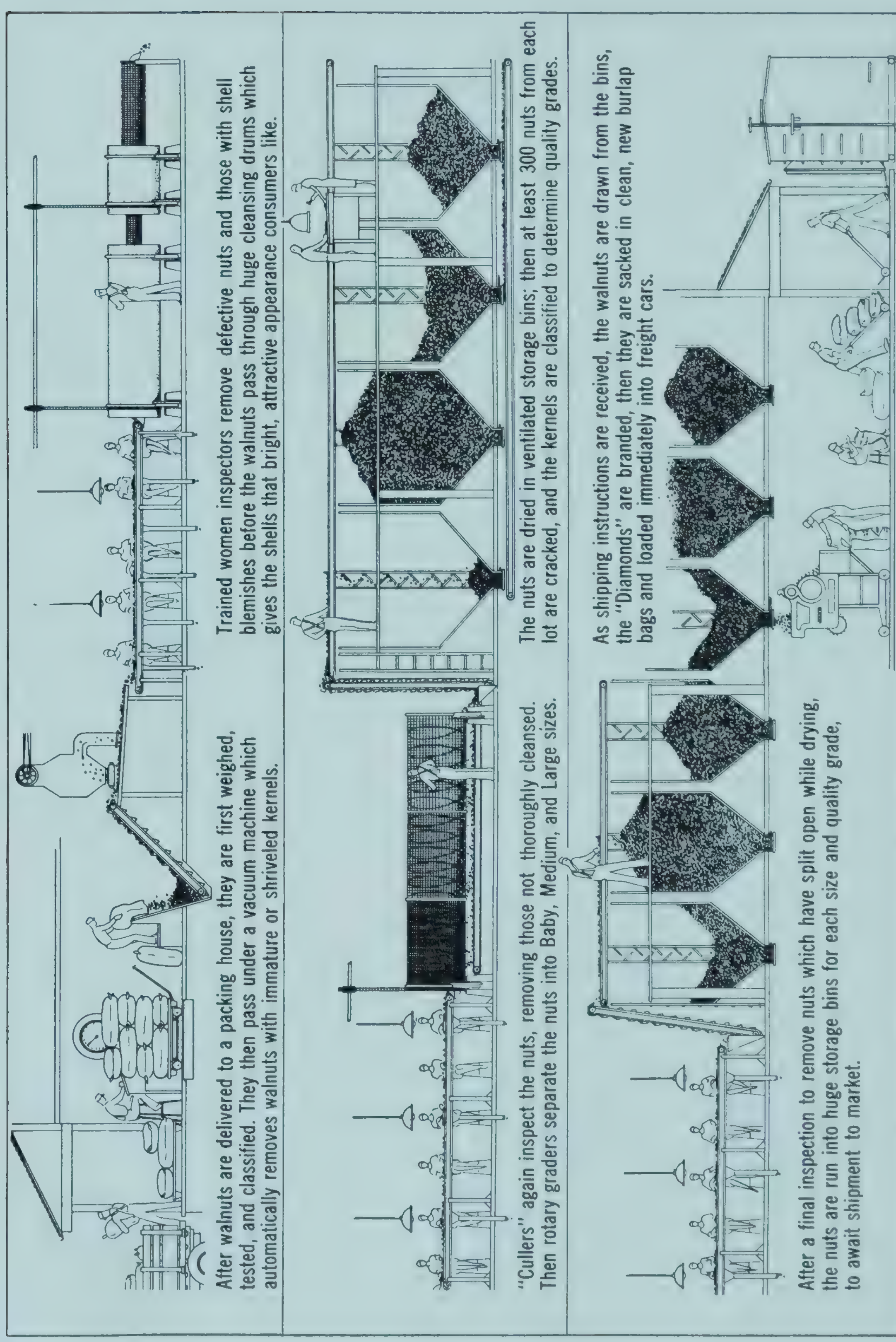


Fig. 15.3. A pictorial operation process chart of English walnut processing.

Fig. 15.3 is a pictorial operation process chart of English walnut processing. Charts of this type are more difficult to make than the graphic type, but they are frequently used for popular presentation because of clarity. Data can be added so that an analysis can be made if needed.

15.6. Flow-Process Charts. Fig. 15.4 is a flow process chart of the portion of the operation process chart in Fig. 15.1 designated as (*b*), (*d*), and (*e*). A detailed study is desired in order to correct the delay prior to the treating operation.

The flow-process chart form is so set up that columns are provided for most of the information required in any study. The blank columns can be used as the situation demands, in this case, for labor and power.

Relative to the information on the chart, consider the following. Each individual step is listed even though some steps may be of minor importance. The chart symbols and identification marks are the same as used in the operation process chart.

If single units are moved by hand, the distance moved may be of significant value because of the labor used. If movement is by mechanical conveyor, the distance may not materially influence labor requirement. A "V" following a distance indicates that it is vertical. The distance index for conveying to delay chamber indicates a total travel of 25 ft, there being a vertical lift of 10 ft. The distance moved is horizontal if no identifying index is used. Information concerning the direction and method of movement may be used for determining materials-handling design data.

The unit times will probably be important if distinct units of material are being processed, animals in an abattoir, small lots of vegetables in a community canning plant, a kettle of sorghum, or any batch process, for example. In cases of this kind, the time of each step is important since the arithmetical total is the total time required and the breakdown indicates steps that might be shortened. The unit times for continuous processes such as vegetable washers, seed cleaners, drum dehydrators, and walnut hullers are not so important. Since the events occur simultaneously, labor is usually not proportional to the individual times and inspections; delays and storages can occur without affecting the normal flow; the total of the times listed is *not* usually the total time required for a complete process. However, the total of the unit operation times and the unit transportation times is the total time

required for the actual processing. The individual values are important in determining the time the material is subjected to an

FLOW PROCESS CHART

Subject charted Seed Cleaning and Treating	Sheet 1 of 1 sheets
Chart begins Material Entering Primary Separator *(b)*	Date 7-6-53
Chart ends Material in Storage *(d–e)*	Charted by M. J.
Type of chart Material	O Operation ⇨ Transportation
Amount or rate charted 50 bu/hr	□ Inspection D Delay ▽ Storage

Process Description of Present Method	Chart Symbols	Dist. Moved	Unit Times in hr Oper.	Insp.	Tran.	Delay	Stor.	Man-hr	Power, (kwhr)			
Primary separation	1		0.67					0.1	4.25			
Inspect for foreign material	1			0.02				0.1				
Convey to gravity table	1 C	12′ v			0.01							
Final separation	2		0.67					0.2	3.20			
Inspect for quality	2			1 wk				0.3				
Convey to delay chamber	2 E	25′ (10 v)			0.01				0.90			
Delay to treat	1					0.33						
Seed treated	3		1.00					0.2				
Convey to sacker	3 E	8′			0.01				0.75			
Sacked	4		1.00					0.7				
Sacks conveyed to storage	4 T	60′			0.01			0.6				
Storage for sale	1						5 mo					
Totals		105′						2.2	9.10			
Note: Under transportation, C is by chute, E by elevator, T by truck, v indicates vertical distance.												

Fig. 15.4. A flow process chart.

event, whether certain operations are performing at peak capacity, and, eventually, what can be done to adjust difficulties due to rate of movement.

The blank spaces are provided for tabulating important data such as labor, utilities, moisture contents, grade factors, tempera-

tures, space requirements, and other information needed to solve a specific problem.

Regarding the specific example of Fig. 15.4, the delay is due to low capacity of the treater. Note that if the capacity of the treater were doubled, the delay could be eliminated and the rate increased to that of the gravity table. In addition, the capacity of the primary separator and sacker would be increased to more satisfactory levels. Two-and-two-tenths man-hours of labor are now required; consequently, three men must be on duty. Doubling the capacity would not affect the labor requirement of any operation except sacking and conveying sacks to storage. They would increase some but not enough to require more than the three men now on duty. Consequently, a second or larger treater could be used to double the capacity and would eliminate the delay without increasing the labor or power requirement.

15.7. Layout Diagram. The layout or flow diagram is used to locate or rearrange the processing units and to route the material through the plant in the most efficient manner.

Before preparing the layout or flow diagram, the following information should be secured.

A. Relative to the entire process, refer to operation and flow process charts.
   1. Rate and characteristics of flow in each line. If the rate varies, or if flow is intermittent, the maximum and minimum rates, their frequency, duration, and time of occurrence should be ascertained. This information is necessary for design of storage and holding areas and surge chambers as well as for determining the optimum size of processing units.
   2. Characteristics of structure and area in which the process is or is to be carried out.
      a. Service facilities, roads, railroads, their capacity, connections, relative location, etc.
      b. Available utilities and quantities.
      c. Orientation and prevailing wind characteristics: operations should be so orientated that workers will not be required to face the sun or to work in heat, dirt, or odors forced upon them by the wind.

d. Details of structure and area: a plan drawing should be prepared showing all the details important from the standpoint of the processing operation. This will be used for the layout diagram. If available, the plan sheet of the blueprints can be used admirably.

B. Relative to each step listed on the process charts.
1. Capacity of unit: this might be a number of small units with accumulated capacity.
2. Labor requirement and characteristics: space needed for workers, light, etc.
3. Space requirements for unit, service, and labor.
4. Utilities required: light, heat, refrigeration, power, water, steam, gas, sewage, ventilation, etc.
5. Materials required for flow in and out of unit.

A new layout can be prepared or an existing layout can be criticized on the basis of the above information. Engineering judgment must be exercised in this step since no two plants are the same. However, the following suggestions, which are based upon a discussion of plant layout practices by Maynard and Stegemerten [4] will be of measurable assistance in most layout jobs.

15.8. Layout Procedures. Many plants have been and some still are laid out on the basis of the following principles.

1. The raw product should come in at one end of the plant and should emerge in the finished state at the other end.
2. Aisles should be provided for transportation purposes and should be kept clear at all times.
3. Like operations should be grouped and arranged in straight lines or orderly rows.
4. Ample space for placing material should be provided around each operation.

Layouts prepared on the basis of these principles were pleasing in appearance, and orderliness and lack of crowding resulted. However, after making a number of detailed studies of this type of setup, many inefficiencies were found to exist. Material and employees had to travel too far. Materials-handling labor was high, it was frequently difficult to provide an efficient flow pattern for the material, and there was too much waste space. This

realization and careful studies produced a new set of principles which are:

1. The material from one operation should be placed in such a position that it can be most easily picked up for the next operation.
2. The distance an operator must move to obtain or deliver material away from a machine should be reduced to a minimum.
3. Time spent by a machine performing an operation or part of an operation which does not require the immediate attention of the operator or attendant is idle time as far as the attendant or operator is concerned.

Plants laid out on these three principles are very efficient, although the arrangement may appear chaotic to the uninformed.

The layout can be most efficiently studied by using a floor plan of the area to be utilized and templets of the equipment to be located or rearranged. The templets should represent the equipment to scale and can be cut out of heavy paper, cardboard, plywood, or any suitable material. Three-dimensional templets may be advisable if height is a factor. Three-dimensional templets can be made of heavy paper, cardboard, or cut from wood blocks. The templets can be shifted to various locations, and studied and criticized until the most satisfactory arrangement is found. The templets can be fastened with thumbtacks, map tacks, rubber cement, staples or any other suitable fastening. String, perhaps colored, can be used to show the direction of flow.

REFERENCES

1. *A.S.M.E. Operation and Flow Process Charts.* American Society Mechanical Engineers. 1947.
2. Barnes, R. M. *Motion and Time Study.* Wiley. Third Edition. 1949.
3. Barnes, R. M. *Works Methods Manual.* Wiley. 1944.
4. Maynard, H. B., and Stegemerten, G. J. *Operation Analysis.* McGraw-Hill. 1939.
5. Mundel, M. E. *Systematic Motion and Time Study.* Prentice-Hall. 1947.

CHAPTER 16

Manual Operation Economy

Processing operations are automatic, semiautomatic, or manual. Automatic operations do not require continuous attendance, for example, refrigeration, drying hay with forced air, separation of cream, etc. Operations that require continuous attendance of an operator whose main function is supervision or control would be considered semiautomatic, for example, evaporating cane syrup to molasses, operating a holding pasteurizer, rendering lard, etc. Processing farm products includes many operations that are a direct function of manual activity, such as candling eggs, grading fruit, dressing poultry, feeding a hemp mill, and packing fresh vegetables for shipment.

The cost of a manual operation is directly proportional to the time required to do it. A competent worker is capable of a definite amount of manual output per day under good conditions. His surroundings should be so managed and arranged that the maximum amount of his available energy will be expended usefully. Management must cooperate in the following respects in order to insure itself economy in manual operation.

1. In order that labor may have the desire to produce, management must provide the following:
   a. Satisfactory working conditions: good light, proper temperature, clean surroundings, good service facilities.
   b. Proper personnel relationships and techniques: Foremen and supervisors must know and use proper managerial methods; work and rest periods must be properly balanced.
2. Equipment layout must permit labor to produce at maximum rate with minimum effort.

ıt *a* under number one was discussed indirectly in Chap. 14. *b* is beyond the scope of this book, but the student must

realize its importance and become versed in the implied principles. Point two will be discussed briefly, although it will be impossible to treat it in as comprehensive a manner as the subject needs.

Motion and time study is a distinct field that is too involved and specialized to be treated completely in this short chapter. The discussion that follows will help solve many of the simpler problems that result because of improper use of motion by labor, but major problems should be handled by experienced motion and time-study men or on the basis of material from the references at the end of this chapter.

16.1. Definition of Motion and Time Study. Barnes[1] has defined motion and time study thus:

> "Motion and time study is the analysis of the methods, of the materials, and of the tools and equipment used, or to be used, in the performance of a piece of work—an analysis carried on with the purpose of (1) finding the most economical way of doing this work; (2) standardizing the methods, materials, tools, and equipment; (3) accurately determining the time required by a qualified person working at a normal pace to do the task; and (4) assisting in training the worker in the new method."

The four parts of the field are distinct divisions of study, but the solution of any specific problem cannot be made on any single one. They must be used in combination.

16.2. Work Economy Principles. Barnes[1] has listed ten principles of motion economy which, when followed, permit the worker to do a maximum amount of work with a minimum of effort. These principles follow.

1. *Motions of the two hands should be simultaneous and symmetrical.*

Unless special thought and study have been given a manual job, most jobs will be performed in some manner such as this. The left hand will pick up a part and hold it while the right hand performs some job upon it. After finishing the work, the left hand rejects the finished job and the process is repeated. Actually, the left hand assists the right hand and the rate of production is a function of the right hand's activity. If both hands are trained and permitted to perform simultaneously, the rate of production will be increased without increasing the physical output of the operator.

This principle is particularly related to fruit and vegetable picking operations. Two-handed symmetrical production is approximately 20 per cent faster than one-handed operations in which one hand assists the other.

The pin board of Fig. 16.1 can be used to demonstrate the importance of this principle. The pins are placed in front of the

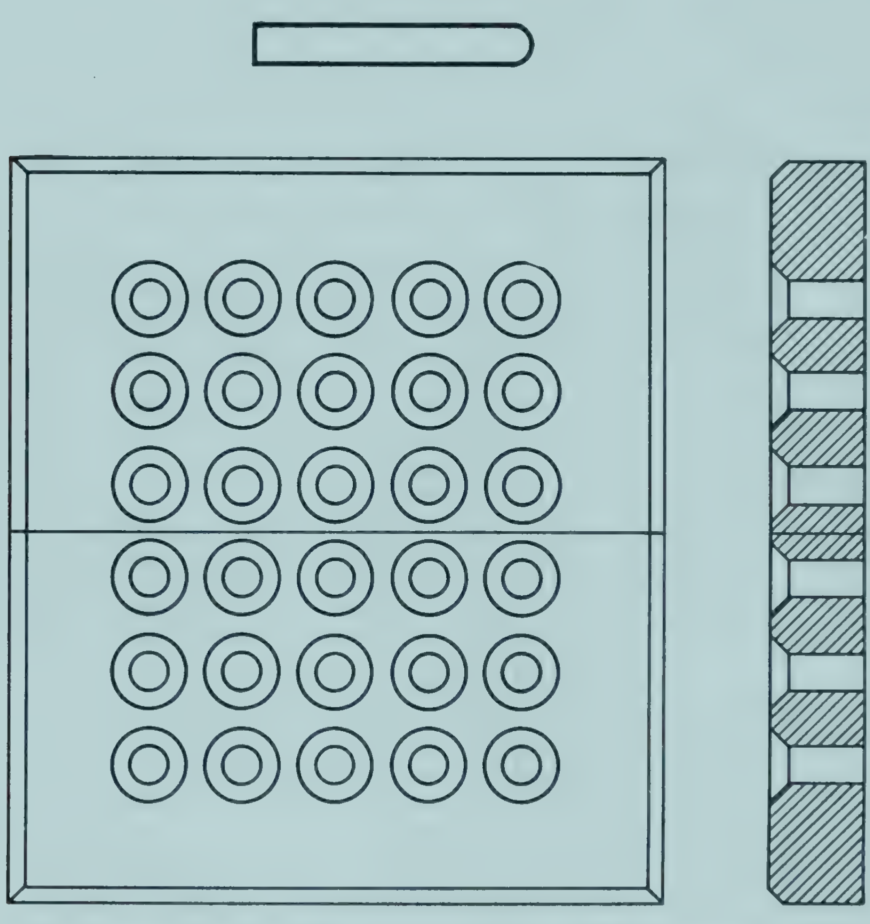


Fig. 16.1. Pin board for work economy study. The board is 7 in. by 8 in. with 3/8 in. holes 1 in. on center and countersunk to 3/4 in. diameter. Pins are 3/8 in. by 3 in. long.

board in a random pile. The time required to fill the board is noted for the one-handed procedure, that is, one hand passing the pins to the other hand which places them curved end down in the board. The time is then observed for two-handed simultaneous independent movements. Barnes has found the average times to be 38 and 23 sec respectively.

2. *Tools and materials should be located close in and directly in front of the operator so as to be within easy reach of the hands. Transport distances should be as short as possible and movements should be as few as possible.*

The working space of an operator is that defined by his hands when his extended arms are moved in three planes without moving his shoulders. The best space is that directly in front of the operator and which is described by both hands.

This principle can be demonstrated by the pin board. Separate the board into two parts as shown and move them 12 in. apart. Observe the time for a simultaneous two-handed fill. Repeat with a 24-in. spacing. Average times will be approximately 26 and 30 sec as against 23 sec for the time with blocks adjacent.

3. *There should be a definite and fixed place for all tools and materials.*

The reason for this is evident. Automatic, fast production movements cannot be developed if the operator has to search with his eyes and perhaps hands before making a productive motion.

4. *The material should be delivered close to the point of use by conveyors or gravity.*

The material should be placed within the space indicated under principle 2. The location of the material should not move. The operator should not be required to use his eyes to guide his hands when reaching for material. The rotating lemon bin of Fig. 16.2 has a movable spring-supported bottom which maintains the lemons at a constant level as regards the operator. This facilitates reaching for fruit that is to be packed which in turn permits a higher rate of packing than would be possible if the lemon level varied.

5. *Tools and materials should be pre-positioned wherever possible.*

Materials, tools, and equipment should be so positioned that a minimum of physical movement is required to perform an operation. The butcher shop with the saws and cleavers hanging above the meat block and knives sheathed to the block all within easy reach of the butcher is a good example of this principle of operation. The handles of all the tools are in such a position that the hand does not have to move far or the wrist twist much to grasp a tool.

6. *"Drop deliveries" should be used wherever possible.*

Essentially, drop delivery implies that upon completion of an operation, the product is released and drops into a box, chute, or conveyor. This type of delivery is very economical of time, arm

movement, and energy. Fig. 16.3 is a drop delivery for reject lemons. Note the short distance and small amount of lifting required to complete an operation. This contributes to both efficiency and speed.

7. *The hands should be relieved of all work that can be done by the feet, power-operated tools, and jigs and fixtures.*

Fig. 16.2. The lemon pack-out bin has a spring-supported bottom, which maintains the top position at the same elevation irrespective of the quantity in the bin. The bin rotates slowly, therefore filling is uniform. (*Courtesy* Food Machinery Corp.)

The trash can with a foot-operated lid minimizes the time and energy required to dispose of a quantity of refuse. In a feed-mixing plant an electric sack sewer which is moved across the sack by hand is faster and less tiring than doing the job by hand with a hand shuttle.

8. *Materials and equipment should be located to permit the best sequence of motions. Rhythm is essential to a smooth easy work pattern.*

Rhythm implies that motions in a productive sequence occur systematically without distinct breaks or abrupt changes and

proceed with a characteristic swing. Circus roustabouts driving tent stakes, an experienced farmer shoveling grain, an experienced egg candler are examples of the type of rhythm desired.

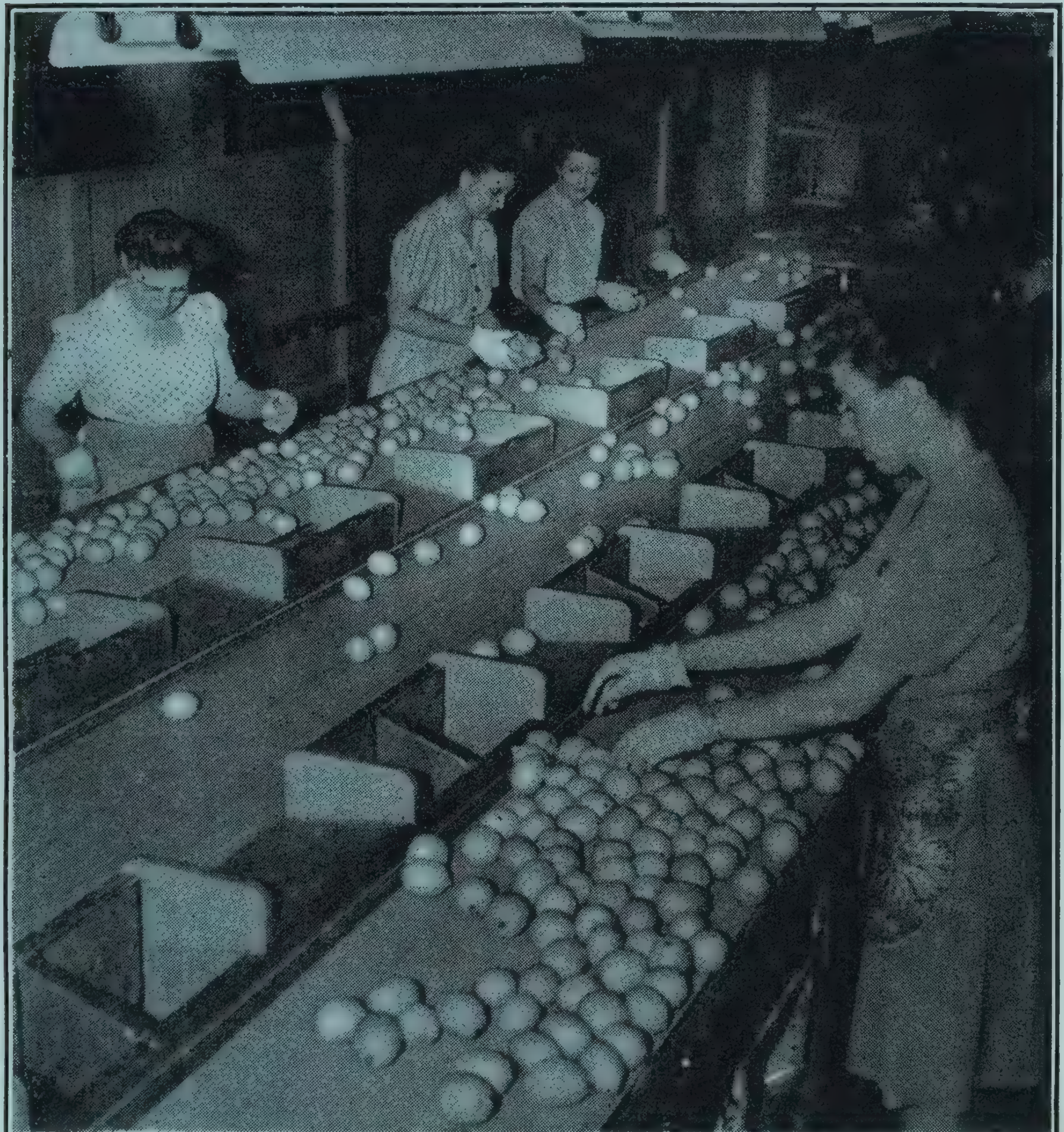


Fig. 16.3. Sorting effort is reduced when drops are used to receive the rejects. Note the short distance the hand must move to reject an undesirable fruit. (*Courtesy* Food Machinery Corp.)

9. *Smooth continuous motions of the hands are preferable to zigzag motions or straight-line motions involving sudden and sharp changes in direction.*

This principle is directly related to principle eight. Studies have shown that in such a simple operation as moving a pencil back and forth across a sheet of paper, that 15 to 25 per cent of

the time is consumed in changing direction of motion of the hand. Comparable inefficiencies exist on processing operations that require zigzag or jerky movements.

10. *The height of the work place and the chair should preferably be arranged to permit alternate sitting and standing at work. Adequate lighting should be provided, and the worker should be made as comfortable as possible.*

Most work positions can be designed so that the operator can sit or stand as desired. This contributes to efficiency. Lighting and other surrounding conditions must conform to recognized industrial standards.

16.3. Standardization. After methods have been developed which meet a specific objective, management must (1) provide facilities, (2) train the workers in the accepted procedures, (3) insure maintenance of the standard, and (4) do these things in the easiest and most economical manner.

Standardized methods involve machines, men, materials, and surroundings. A record should be made indicating all the details that are necessary to carry out the accepted procedure. This record should indicate the design of the machines, their operating characteristics, the specifications of materials, and working conditions. A procedure should be developed for checking and maintaining these conditions since machines will get out of adjustment, the characteristic of raw materials will vary from lot to lot, and working conditions may change with time.

Machines and equipment should be standardized as much as possible so that conversion, rearrangement, or change can be made with a minimum of time, labor, and expense. Management must recognize and foresee the possibility of variation in production and provide for it. For example, box-making machines should be flexible enough so that they can be changed to make boxes of a different size if needed. Benches and conveyors should be standardized so they can be arranged with a minimum of alteration to handle a different material if material variation is expected.

16.4. Time Study. Time study is the procedure used to evaluate a manual operation or series of unified manual operations in order to provide procedures facilitating the greatest output with average normal worker effort. Three steps are involved in the procedure; these are:

1. *Determine the capacity or normal or standard output of the worker.*

In some cases this can be determined by analyzing the movements of the most productive worker in a group and then using this analysis as a basis for recommendations. Or it may be necessary to analyze and improve the current inefficient procedure to secure a suitable standard.

The standard or basic output or capacity should be based upon the most efficient use of effort possible and should be at such a speed that maximum production can be realized without fatigue to the operator. The standard or minimum wage is usually based upon the standard production rate with incentive piece-work rates for production over and above the standard. This makes it possible for the worker of average energy to do an acceptable amount of work without undue fatigue and receive a satisfactory wage for it. The faster or more energetic worker, at the same time, will be rewarded for his extra production.

2. *Analyze the present steps in the operation to eliminate useless movements and improve necessary movements.*

Frank B. Gilbreth found that there were 18 elemental events used in manual operations. These events or activities, called "therbligs" (Gilbreth spelled backwards), in various combinations compose any and all manual operations or movements, drinking coffee, sharpening a pencil, wrapping a package, picking a chicken, picking apples, to cite a few. The 18 therbligs are listed in Table 16.1.

After dividing an operation into the therbligs or other significant events that compose it, the objective is to eliminate the unnecessary therbligs, combine as many of the necessary therbligs as possible, and then rearrange the remaining therbligs or events in the most satisfactory sequence.

After making the obvious adjustments, the operation is broken down into suitable elements and the elements (which may be composed of more than one therblig) studied on the basis of requisite time as taken with a stop watch. Recent developments in time study with a motion picture camera, called "micromotion" studies, have made possible detailed studies of intricate operations that could not be analyzed by the stop-watch method. Detailed procedures to use will be omitted since they are a separate study in themselves.

Table 16.1 BASIC THERBLIGS

No.	*Therblig*	*Explanation, Suggested by*
1	Search	Eye moving as if searching
2	Find	Eye straight as if fixed on object
3	Select	Reaching for object
4	Grasp	Hand open for grasping object
5	Transport loaded	A hand with something in it
6	Position	Object being placed by hand
7	Assemble	Two or more things put together
8	Use	A manipulation, e.g., turning screw
9	Disassemble	One part of an assembly removed
10	Inspect	Observe, e.g., check for color
11	Pre-position	Placing nail for driving
12	Release load	Dropping content out of hand
13	Transport empty	An empty hand moving
14	Rest for overcoming fatigue	Man seated resting
15	Unavoidable delay	Parts supply exhausted
16	Avoidable delay	Operators visiting
17	Plan	Thought process necessary before operation
18	Hold	Retaining object in hand

16.5. Training the Workers. After an improved procedure has been developed, it must be taught to the operators. This can be done by before-and-after moving pictures, operation charts, diagrams, or demonstrations.

Many industrial plants give all new employees and apprentices instruction in motion economy which includes the work previously discussed plus other information which will help them do their tasks in the easiest and most efficient manner. This training also places them in an excellent position to accept and use new and improved procedures.

REFERENCES

1. Barnes, R. M. *Motion and Time Study.* Wiley. Third Edition. 1949.
2. Barnes, R. M. *Work Methods Manual.* Wiley. 1944.
3. Holmes, W. G. *Applied Time and Motion Study.* Ronald. New York, 1945.
4. Maynard, H. B., and Stegemerten, G. J. *Operation Analysis.* McGraw-Hill. 1939.
5. Mundel, M. E. *Systematic Motion and Time Study.* Prentice-Hall. 1947.
6. Vaughan, L. M., and Hardin, L. S. *Farm Work Simplification.* Wiley. 1949.

Appendix A

SOME CONVERSION FACTORS

Multiply	*By*	*To Obtain*
British thermal units	778.3	Foot-pounds
British thermal units	3.931×10^{-4}	Horsepower-hours
British thermal units	2.930×10^{-4}	Kilowatt-hours
Bushels	1.244	Cubic feet
Cubic feet	7.481	Gallons
Feet of water	0.434	Pounds per square inch
Horsepower	42.40	British thermal units per minute
Horsepower	33,000	Foot-pounds per minute
Horsepower	745.7	Watts
Horsepower (boiler)	33,475	British thermal units per hour
Horsepower-hours	2,544	British thermal units
Inches of water	0.00246	Atmosphere
Inches of water	69.6	Feet of air
Inches of water	0.0736	Inches of mercury
Inches of water	0.03613	Pounds per square inch
Kilowatt-hours	3,413	British thermal units
Pounds	7,000	Grains
Pounds	453.6	Grams

Appendix B

BULK DENSITIES *

(From *Agricultural Statistics*, U. S. Department of Agriculture, 1952)

Material	*Lb per cu ft*
Alfalfa seed	48.0
Apples	38.4
Barley	38.4
Beans	48.0
Bluegrass seed	11.2–24.0
Castor beans	36.8
Corn, ear	28.0 †
Corn, shelled	44.8
Grain sorghums	44.8 and 40.0
Rice, rough	36.0
Rye	44.8
Soybeans	48.0
Sweet potatoes	44.0
Walnuts	40.0
Wheat	48.0

* As processed for commercial channels, grains at approximately 14 per cent moisture content (wet basis).

† Two cubic feet of ear corn will produce approximately 1 cu ft of shelled corn.

Appendix C

HEATS OF COMBUSTION AND WEIGHTS PER GALLON

Fuel	Heat Value, Btu per Lb High	Heat Value, Btu per Lb Low	Weight per Gallon, lb
Gasoline	20,750	19,500	6.15
Kerosene	19,810	18,500	6.82
Fuel oil (Calif.)	18,835	17,755	7.96
Butane	21,180	19,660	4.84
Propane	21,560	19,930	4.24
Methane (natural gas)	995.4 *	896.2 *	——
Coal (Pa.)	12,500	——	——

1 kwh = 3413 Btu; 1 hp-hr = 2544 Btu.

* Per cubic foot at 68° and 14.7 lb per sq in. abs.

Appendix D

AVERAGE SPECIFIC HEATS BETWEEN 32 AND 212°F

	Specific Heat
Solids:	
Concrete	0.25
Brick	0.20
Wood (pine)	0.45
Wood (oak)	0.55
Plaster	0.20
Steel	0.12
Brass	0.089
Nickel steel	0.109
Stainless steel	0.117
Asbestos	0.20
Cork	0.485
Glass (flint)	0.12
Glass (Pyrex)	0.20
Ice	0.505
Humus (soil)	0.44
Copper	0.095
Liquids:	
Water	1.00
Milk (whole)	0.93
Ammonia	1.10
Freon (F-12)	0.25
Mercury	0.033
Petroleum	0.511
Olive oil	0.471
Gases (and Vapors):	
Air	0.24
Ammonia	0.52
Freon	0.143
Nitrogen	0.244
Water vapor	0.47
Hydrogen	3.40
Farm Products:	
String beans	0.87
Lima beans	0.75
Dry beans	0.30
Cantaloupe	0.91
Corn (green)	0.86
Onions	0.91
Potatoes (white, sweet)	0.86
Bacon	0.50
Beef	0.75
Lamb	0.67
Pork	0.68
Poultry	0.79
Fruits (fresh)	0.88–0.92
Butter	0.64
Eggs	0.76
Nuts	0.21–0.29
Wheat	0.39
Soybeans	0.47

Appendix E

SOME PIPE AND TUBING DIMENSIONS

Standard Iron and Steel Pipe *

Nominal Size, in.	*Actual OD, in.*	*Approximate ID, in.*
$\frac{1}{8}$	0.405	0.27
$\frac{1}{4}$	0.540	0.36
$\frac{3}{8}$	0.675	0.49
$\frac{1}{2}$	0.840	0.62
$\frac{3}{4}$	1.050	0.82
1	1.315	1.05
$1\frac{1}{4}$	1.660	1.38
$1\frac{1}{2}$	1.900	1.61
2	2.375	2.07
$2\frac{1}{2}$	2.875	2.47
3	3.500	3.07

Standard Copper Tubing †

Size, OD, in.	*Wall Thickness, in.*
$\frac{1}{8}$	0.030
$\frac{3}{16}$	0.030
$\frac{1}{4}$	0.030
$\frac{5}{16}$	0.032
$\frac{3}{8}$	0.032
$\frac{1}{2}$	0.032
$\frac{5}{8}$	0.035
$\frac{3}{4}$	0.035

Stainless-Steel Tubing ‡

Size, OD, in.	*Wall Thickness Range, in.*
$\frac{1}{8}$	0.004–0.049
$\frac{1}{4}$	0.004–0.083
$\frac{3}{8}$	0.005–0.134
$\frac{1}{2}$	0.009–0.165
$\frac{3}{4}$	0.014–0.238
1	0.018–0.313
$1\frac{1}{4}$	0.022–0.313
$1\frac{1}{2}$	0.025–0.375
$1\frac{3}{4}$	0.028–0.500
2	0.032–0.625
$2\frac{1}{2}$	0.035–0.750
3	0.035–1.000

* Extra strength pipe with the same OD but with reduced ID is available in a number of ranges.

† Various wall thicknesses, the OD remaining constant, are available. Tube types K, L, M for compression and soldered fittings are $\frac{1}{8}$ in. larger in OD than the standard tubing (e.g. a $\frac{1}{2}$ in. type K would be $\frac{5}{8}$ in. in OD).

‡ Stainless-steel pipe of the same dimension as iron pipe is available.

Appendix F

STEAM TABLE *

°F	Absolute Pressure, lb per sq in.	Specific Volume, cu ft per lb: Saturated Liquid	Specific Volume, cu ft per lb: Saturated Vapor	Enthalpy, Btu per lb: Saturated Liquid	Enthalpy, Btu per lb: Saturated Vapor	Entropy, Btu per °F lb: Saturated Liquid	Entropy, Btu per °F lb: Saturated Vapor
32	0.08854	0.01602	3306	0.00	1075.8	0.00	2.1877
35	0.09995	0.01602	2947	3.02	1077.1	0.0061	2.1770
40	0.12170	0.01602	2444	8.05	1079.2	0.0162	2.1597
45	0.14752	0.01602	2036.4	13.06	1081.5	0.0262	2.1429
50	0.17811	0.01603	1703.2	18.07	1083.7	0.0361	2.1264
55	0.2141	0.01603	1430.7	23.07	1085.8	0.0459	2.1104
60	0.2563	0.01604	1206.7	28.06	1088.0	0.0555	2.0948
65	0.3056	0.01605	1021.4	33.05	1090.2	0.0651	2.0796
70	0.3631	0.01606	867.9	38.04	1092.3	0.0745	2.0647
75	0.4298	0.01607	740.0	43.03	1094.5	0.0839	2.0502
80	0.5069	0.01608	633.1	48.02	1096.6	0.0932	2.0360
85	0.5959	0.01609	543.5	53.00	1098.8	0.1024	2.0222
90	0.6982	0.01610	468.0	57.99	1100.9	0.1115	2.0087
95	0.8153	0.01612	404.3	62.98	1103.1	0.1205	1.9955
100	0.9492	0.01613	350.4	67.97	1105.2	0.1295	1.9826
110	1.2748	0.01617	265.4	77.94	1109.5	0.1471	1.9577
120	1.6924	0.01620	203.27	87.92	1113.7	0.1645	1.9339
130	2.2225	0.01625	157.34	97.90	1117.9	0.1816	1.9112
140	2.8886	0.01629	123.01	107.69	1122.0	0.1984	1.8894
150	3.718	0.01634	97.07	117.89	1126.1	0.2149	1.8685
160	4.741	0.01639	77.29	127.89	1130.2	0.2311	1.8485
170	5.992	0.01645	62.06	137.90	1134.2	0.2472	1.8293
180	7.510	0.01651	50.23	147.92	1138.1	0.2630	1.8109
190	9.339	0.01657	40.96	157.95	1142.0	0.2785	1.7932
200	11.526	0.01663	33.64	167.99	1145.9	0.2938	1.7762
210	14.123	0.01670	27.82	178.05	1149.7	0.3090	1.7598
212	14.696	0.01672	26.80	180.07	1150.4	0.3120	1.7566
220	17.186	0.01677	23.15	188.13	1153.4	0.3239	1.7440
230	20.780	0.01684	19.38	198.23	1157.0	0.3387	1.7288
240	24.969	0.01692	16.323	208.34	1160.5	0.3531	1.7140
250	29.825	0.01700	13.821	218.48	1164.0	0.3675	1.6998
260	35.429	0.01709	11.763	228.64	1167.3	0.3817	1.6860
270	41.858	0.01717	10.061	238.84	1170.6	0.3958	1.6727
280	49.203	0.01726	8.645	249.06	1173.8	0.4096	1.6597
290	59.556	0.01735	7.461	259.31	1176.8	0.4234	1.6472
300	67.013	0.01745	6.466	269.59	1179.7	0.4369	1.6350

* Abstracted from: Keenan and Keyes, *Thermodynamic Properties of Steam*, Wiley, 1936.

Index